HANDBOOK

OF

CARBURETION

BY

ARTHUR BENJ. BROWNE

Consulting Engineer

Member Society of Automobile Engineers

Member American Institute of Mining Engineers

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FOREWORD

CONSENSUS of public opinion, both technical and lay, would undoubtedly be singularly unanimous in welcoming the complete abolition of the carbureter. No part of a motor-vehicle is less understood or more abused, in thought and deed. No other part of the entire mechanism of the car is subjected to the indignities that are heaped upon the carbureter.

This condition will continue to exist until the genius which has already made such colossal strides in automobile engineering turns its serious attention to an understanding of the fundamental laws governing carburetion.

The development of any branch of science depends largely on the recognition of fundamental principles. These principles may be as accurately determinative as are those expressed in Ohm's law, or as purely theoretical as is Dalton's Atomic Theory, but the science of chemistry surely owes no less to Dalton than the electrical field owes to Ohm.

In the science of carburetion it is difficult, perhaps impossible, to secure practical measurements as definite as those from which Ohm deduced his law, but the application of natural laws to problems of carburetion is so far less an excursion into the realms of pure theory than that by which Dalton revolutionized science, that its recognition and universal acceptance by practical men seems overlong delayed. Still, the lack of uniformity exhibited by the great and ever-increasing variety of carbureters on the market proves that, as yet, no comprehensive principle of automatic regulation of the gas to air ratio has been generally recognized.

The simplest form of carbureting device consists of a fuel jet introduced into the moving air column within the intake pipe. If the velocity of the fuel flow were directly proportional to the velocity of the air flow, the mixture from such a device would be

of constant composition under all conditions and the principal problem of carburetion would be resolved at once to its simplest terms. Unfortunately, the relation between the air and fuel velocities is not a direct proportion, but, as will be demonstrated, it is none the less definite. Once recognized, its application to practical carburetion not only eliminates the necessity for most of the mechanical complications now in use, but it explains clearly the errors which are introduced by their use.

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HANDBOOK OF CARBURETION

CHAPTER I

THEORY OF CARBURETION

THE law of the flow of fluids, including gases within certain limits of pressure differences, is expressed

$$v = \sqrt{2gh} \qquad (1)$$

where

v = velocity in feet per second.
g = acceleration of gravity (32.2 feet per second).
h = head, or height in feet of the fluid, required to produce the pressure necessary to cause the flow.

The velocity of the air (Va) in a carbureter will be expressed

$$Va = \sqrt{2gh} \qquad (2)$$

whence

$$(Va)^2 = 2gh \qquad (3)$$

and

$$h = \frac{(Va)^2}{2g} \qquad (4)$$

In this case, h is the height in feet of a column of air, the weight of which will exert the pressure necessary to cause a flow of air at the velocity Va, or conversely, the loss of head caused by the air flowing at the velocity Va.

The value of h, or as applied to carburetion h', must not be understood to be literally the vertical measurement between the surface of the fuel in the float reservoir and the mouth of the fuel nozzle. To this must be added the "friction head" imposed on the fuel by its passage through the nozzle. This is subject

to constant variation and depends in value upon the velocity, density, and viscosity of the fuel. The exact value of h' is probably indeterminable, and so, for use in the following illustrative formulæ, it will be assigned the numerical value of the vertical distance, without attempt at correction.

The head of fuel caused by air passing at a velocity Va will be

$$h \times \frac{Wa}{Wf}$$

where

Wa = weight 1 cubic foot of air (.076 pounds at 62° F.).
Wf = weight 1 cubic foot of fuel (weight 1 cubic foot of water $\{62.355\}$ pounds × sp. gr. of fuel).

Applying equation (2) to the fuel velocity, Vf, we have

$$Vf = \sqrt{2gh\frac{Wa}{Wf}}$$

But as, before actual discharge commences, the fuel must rise from the level in the float chamber to the mouth of the fuel nozzle, a distance of h' feet, subject to the retardation of gravity, we must deduct the value of $2gh'$, and hence

$$Vf = \sqrt{2gh\frac{Wa}{Wf} - 2gh'}$$

Substituting the value of $2gh$ as determined by equation (3), the velocity of the fuel is expressed in terms of air velocity as follows:

$$Vf = \sqrt{\frac{Wa}{Wf}\,Va^2 - 2gh'} \tag{5}$$

Wimperis ("The Internal Combustion Engine," page 268) arrives at the same relation between air and fuel velocities by methods of the calculus.

Effect of Temperature

The density of both fuel and air is, of course, modified by temperature. The density of the air varies inversely as the

absolute temperature, while the density of gasoline is shown by Clerk and Burls ("The Gas, Petrol, and Oil Engine," Vol. II, page 623) to be modified by temperature as follows:

$$\text{Sp. gr.} = 0.72 \left\{1 - .0007\,(t - 60)\right\}$$

whence

$$Wf = W \times s \left\{1 - .0007\,(t - 60)\right\} \tag{6}$$

where

Wf = weight of 1 cu. ft. of gasoline.
W = weight of 1 cu. ft. of water.
s = specific gravity of gasoline at 60° F.
t = temperature of the gasoline in F.°
t' = temperature of the air in F.°

Substituting these values in equation (5) we have:

$$Vf = \sqrt{\frac{\frac{(460 + 62)\,0.076}{460 + t'}}{62.355 \times s\left\{1 - .0007\,(t - 60)\right\}} \times (Va^2 - 2gh')} \tag{7}$$

The range of values for t and t' to be used in equation (7) is so small that it will be readily seen that the effect of temperature is negligible.

Working Formulæ

Omitting the temperature correction, a simple working equation for gasoline of a specific gravity of 0.72 may be expressed

$$Vf = \sqrt{(.00169\,Va^2) - 2gh'} \tag{8}$$

For fuel of any other gravity, equation (5) becomes

$$Vf = \sqrt{\left(\frac{.076}{62.355\,s}\,Va^2\right) - 2gh'}$$

which reduces to

$$Vf = \sqrt{\left(\frac{.00122}{s}\,Va^2\right) - 2gh'} \tag{9}$$

Application of the Law to Various Types

In order to obtain a clear understanding of the application of the law, let us consider the action of various types of carbureting devices in view of the relation of air and fuel velocities as expressed in equation (8).

Hypothesis

Assume (A) that a unit quantity of air is passing each device with a given velocity and then (B) that a greater quantity of air is demanded. For the sake of uniformity let us assume that each device maintains a constant level of fuel 0.5 inch (0.0416 feet) below the mouth of the fuel nozzle and that the fuel employed is gasoline of a specific gravity of 0.72.

TYPE I

THE SIMPLE CARBURETER

(A) In this device, the velocity of the fuel discharge for an air velocity of say 90 feet per second will be, by equation (8)

$$Vf = \sqrt{(.00169 \times 90^2) - (64.4 \times .0416 \times 0.72)} = 3.43 \text{ ft. per sec.}$$

(B) As the area of air admission is constant, four times the air will pass at four times the velocity. By equation (8) this will induce a fuel flow of

$$Vf = \sqrt{(.00169 \times 360^2) - 1.9} = 14.73 \text{ ft. per sec.}$$

Tendency Toward Enrichment

Hence, while the quantity of air has been increased four times, the quantity of fuel has increased 4.4 times and the resulting mixture is 10.4 per cent richer than formerly.

TYPE II

THE MIXING VALVE

In this device, head, pressure on the valve, amount of valve opening, admission area exposed by said opening, and the quan-

tity of air admitted are in direct proportion to one another, if friction is disregarded. It follows therefore, that, as the head varies with the square of the velocity (equation 4), the quantity of air bears the same relationship. Conversely we may state that the velocity varies as the square root of the quantity of air admitted.

(A) The fuel flow, for an air velocity of 90 feet per second, will be 3.43 feet per second as in (I-A).

Tendency Toward Impoverishment

(B) By the proportion stated above, four times the initial quantity of air will pass the apparatus at twice the initial velocity. Hence the fuel flow induced by the increased quantity will be, by equation (8)

$$Vf = \sqrt{(.00169 \times 180^2) - 1.9} = 7.27 \text{ ft. per sec.}$$

showing that while the air quantity has increased four times, the fuel quantity has increased only 2.17 times, or but 54 per cent of the fuel is present that is necessary for a constant mixture. It is thus readily seen why the mixing valve cannot be used for carburetion where any material degree of flexibility is desired.

TYPE III

THE COMPENSATING CARBURETER

Attempts to correct the tendency to over-richness exhibited by the simple carbureter led to the early adoption of the auxiliary air-valve. The popular conception of the auxiliary air-inlet is that the air thus admitted serves to dilute the necessarily over-rich mixture formed at the mouth of the fuel nozzle. As all the air entering the carbureter, through either the primary or auxiliary inlet, finally reaches the cylinders as part of the explosive mixture, the foregoing statement is obviously true, but the most important function of the auxiliary inlet is likely to be lost sight of in such an explanation of its purpose.

True Function of the Auxiliary

The area of the auxiliary opening modifies the velocity of all the incoming air and hence exercises a direct influence upon the amount of fuel inspirated. This function will be better understood if the primary and auxiliary inlets are considered as a divided unit. Any enlargement of the auxiliary area increases the *total* area of admission and hence modifies both quantity and velocity.

In a carbureter of this type let

Q = the quantity of air.
V = velocity of the air.
a = auxiliary area.
c = the primary area.
A = total admission area = $a + c$.
g = acceleration of gravity.

Disregarding friction, the quantity of fluid discharged by an orifice is expressed

$$Q = VA \tag{10}$$

Hence the quantity of air passing the carbureter will be

$$Q = V\,(a + c)$$

which, by substituting the value of V from equation (1), may be written

$$Q = (a + c)\,\sqrt{2gh} \tag{11}$$

In this equation h is the height of a column of air necessary to cause a unit deflection of the spring governing the auxiliary valve; therefore the velocity of a given quantity of air is directly dependent upon spring tension and deflection, as well as upon the relative areas of both primary and auxiliary openings. As these variables are fixed by construction, determination of the quantity and velocity may be effected by simple substitution of the known values in equation (11).

For instance, assume that in a carbureter of this type, provided with a primary inlet 5/8 inch in diameter (0.3 square inch area), a vacuum of 1 inch of water causes an auxiliary area of 0.05 square inch to be opened.

(A) A head of 1 inch of water is equivalent to a head of 68.284 feet of air at normal pressure and temperature.

By equation (1)

$V = \sqrt{64.4 \times 68.28} = 66.31$ feet (or 796 inches) per second.

$A = 0.3 + .05 = 0.35$ square inch.

$Q = 796 \times 0.35 = 278.6$ cubic inches per second.

By equation (8)

$Vf = \sqrt{(.00169 \times 66.31^2) - 1.9} = 2.35$ feet per second.

(B) Assume now, that on open throttle, the vacuum within the carbureter is 20 inches of water. The head of air would be $20 \times 68.28 = 1{,}365.6$ feet.

$V = \sqrt{64.4 \times 1{,}365.6} = 296.5$ feet (or 3,559 inches) per second.

$A = 0.3 + (0.5 \times 20) = 1.3$ square inches.

$Q = 3{,}559 \times 1.3 = 4{,}616.7$ cubic inches per second.

$Vf = \sqrt{(.00169 \times 296.5^2) - 1.9} = 12.1$ feet per second.

Tendency Toward Impoverishment

Therefore, the air flow has increased $\frac{4{,}617}{279} = 16.5$ times while the fuel flow has increased only $\frac{12.1}{2.35} = 5.15$ times; or but 31 per cent of the former proportion of fuel is present. In other words, had the original mixture in (A) been in the air/gas ratio of say 10/1, the high-speed mixture of (B) would be in the ratio of 32/1, which is far beyond the limits of combustibility.

Failure of Corrective Devices

As may be readily determined, no adjustment of spring tension can do more than very slightly modify this tendency

toward impoverishment of the mixture, while the addition of various forms of subsidiary springs, becoming operative only at some point of the valve-opening, can do no more than correct the error at one given point and then start, as it were, merely a new scale of errors.

The inherent error of the auxiliary valve is by no means of theoretical interest only. It still remains a factor of so intensely practical effect, despite the remarkable ingenuity that has been displayed in various attempts to correct it, that its elimination would effect an annual saving of thousands of dollars to both manufacturer and user of motor-cars through the increased efficiency of the liquid fuel engine.

TYPE IV

THE MULTIPLE JET CARBURETER

Attempts to correct the error in mixture composition introduced by the increasing air flow have been confined largely to two principal channels. Abroad, the tendency is toward the use of multiple fuel jets, while in this country more attention has perhaps been given to the direct mechanical regulation of the area of the orifice in the fuel nozzle.

Each Succeeding Jet Subject to Error of Type I

It will be apparent from the foregoing treatment of the subject that, in multiple jet practice, the flow from each succeeding jet is, in turn, amenable to the law of fluid flow as expressed in equation (8). Hence, each succeeding jet, like the subsidiary spring on the auxiliary valve of Type III merely corrects the error at the point where its own discharge commences and then the flow suffers a cumulative error until corrected by the introduction of the flow from still another jet.

It is evident that the use of a sufficient number of jets might be made to reduce the error to very small proportions, and in fact good results have been obtained from such construction. Mechanical complications and the nicety of constructional detail have proved serious disadvantages, however.

TYPE V

THE VARIABLE FUEL ORIFICE

Inspection of equation (8) and the substitution of values therein in the examples cited disclose that the fuel velocity is in constantly decreasing proportion to the air velocity. In Type III, the quantity of fuel discharge has been treated of in terms of fuel velocity. It is evident, however, from equation (10) that the actual fuel discharge is the product of its velocity and the area of the fuel orifice. Hence, it will be recognized that variation of the area of fuel orifice may be made to compensate for the increasing ratio between the fuel and air velocities. In III-B, for instance, while the quantity of air was increased 16.5 times, the fuel velocity increased only 5.15 times; therefore, to maintain constancy of mixture, the area of the fuel orifice should have been increased $\frac{16.5}{5.15} = 3.2$ times.

Delicacy of Construction and Adjustment

The withdrawal of a straight tapered pin from the fuel nozzle increases the area of discharge in direct proportion to the lift of the pin; consequently, delicate mechanical complications are resorted to in effecting the desired decrease in the proportional area opened. Properly designed and properly adjusted, there is no reason why this method should not give results approaching accuracy, but when we consider the almost microscopic nicety of adjustment necessary to effect accurate sub-division of the minute fuel stream, we realize the practical difficulty of both making and maintaining such adjustments. When we remember, too, that the volume of liquid gasoline is less than 1/8000 of the volume of the air with which it is mixed, it is apparent that regulation of the 8,000 parts would be much more practical than any attempt to subdivide the one part.

TYPE VI

THE CONSTANT VACUUM PRINCIPLE

If air is admitted to a chamber through an opening which is governed by a weighted valve, a sub-atmospheric pressure or partial vacuum will be maintained in the chamber, equivalent to the weight of the valve per square inch of exposed area.

As the demand for air becomes greater the valve will be lifted higher, admitting just enough air to maintain a vacuum in consonance with the weight of the valve, which is, of course, constant at all times.

Comparison With Type V

The vacuum being constant, it follows that the velocity of the entering air is constant and hence it is necessary to provide some means for increasing the area of the fuel flow. As in Type V, this may be accomplished by withdrawing a tapered needle from the fuel nozzle. In this respect this type has the advantage over Type V, because the increase in fuel flow is a straight line curve and hence the proportional withdrawal of a straight tapered needle maintains constant proportions of flow. The needle may be therefore directly attached to the air-valve, and move with it.

If the taper is properly calculated to allow for decreasing friction as the opening becomes greater, this device should maintain constancy of any given mixture proportions.

Acceleration

During the brief instant when the air-valve is actually being lifted, as on opening the throttle, the vacuum is temporarily increased, because more energy is necessary to *move* the valve than to sustain it in a given position. The result is a slight additional impetus given to the fuel flow. A valve can be designed of such weight that this tendency to increase the richness of the mixture is nearly, or quite, counter-balanced by the inertia of the fuel. Such a design gives prompt and very satisfactory acceleration.

Constancy cannot be maintained, however, when the mixture proportions are to be varied by adjusting the needle relative to the nozzle, because, as the areas of circles vary as the square of their diameters, the annulus between the needle and the nozzle at minimum opening is directly proportional to the corresponding annulus at full opening, only when the needle is in the position for which the areas were determined.

Advantages

This type presents the distinct advantages of:

First.—Relatively high velocities at low speeds, insuring comparative ease of starting, and making slow running possible either on full or part throttle.

Second.—No increase of velocity at extreme high speeds, hence no reduction of volumetric efficiency.

Third.—Comparative freedom from the effects of barometric changes.

TYPE VII

THE COMPENSATING NOZZLE

As has been shown in Type I, the tendency of a simple fuel jet in a moving air column is toward enrichment. Attempts have been made to correct this tendency by using a second nozzle, which receives a limited flow of fuel from an orifice of such area that, as the air velocity increases, insufficient fuel is delivered and the resulting mixture becomes leaner. By combining the two nozzles, the first with its tendency to enrichment and the second with its tendency to impoverishment, it is claimed that a balance is established which produces a constant mixture.

Difference in Governing Laws

The discharge from the enriching nozzle follows the law of fluid flow, while the action of the compensating nozzle is dependent solely on the friction on the fuel in passing into the nozzle. It is doubtful, therefore, if the reaction between the two nozzles is more than an approximation to true compensation.

Instruments of this type are widely used in European practice, and to a considerable extent in this country. Their freedom from moving parts is attractive, and their performance is as good as that of many other types.

Close Adjustment Necessary

They are subject, however, to several disadvantages:

First.—Because the compensation is effected by friction, any approach to accuracy is confined within comparatively narrow limits of air quantities, and is even then obtained only with minute accuracy of workmanship and final adjustment.

Additional Starting Device Required

Second.—The air for all speeds is admitted through a single opening of fixed area. This area must be sufficiently large to prevent undue friction, or wire-drawing at high speeds. In consequence, it must be too large to insure proper atomizing velocities at starting speeds. The latter is commonly provided for by a third jet inserted near the edge of the butterfly throttle. In effect, this is a separate carbureter, operative only when the throttle is nearly closed.

Action on Full Throttle

Third.—Because of the fixed area of the air-inlet, it is to be expected that flexibility on open throttle will be sacrificed. Either maximum speed will be curtailed by wire-drawing, or the engine will not run slowly under heavy load and full throttle opening.

Acceleration

Fourth.—As the density of fuel is greater than that of air so is its inertia increased. As a result, upon suddenly opening the throttle for acceleration, the mixture is momentarily impoverished as the air flow exceeds that of the fuel. The result is that this type does not give that instant response to the throttle that is desirable.

TYPE VIII

COMPENSATION BY VELOCITIES

It has been shown in Type V that compensation can be effected by the variation of the area of the fuel nozzle. It is equally true that automatic variation of the total air admission area will accomplish the same result with much greater accuracy and without adjustments or mechanical complications of any kind. For this purpose it is necessary to determine the velocity of the air corresponding to any given fuel velocity.

If, by equation (8)

$$Vf = \sqrt{(.00169\ Va^2) - 2gh'}$$

$$Va = \sqrt{\frac{1}{.00169}(Vf^2 + 2gh')}$$

or, more conveniently,

$$Va = \sqrt{591.71\ (Vf^2 + 2gh')} \tag{12}$$

The practical application of these formulæ is, perhaps, best made clear by a concrete example. Let us consider a carbureter with a primary inlet $\frac{5}{16}$ inch in diameter (area, 0.077 square inch). Let us assume the auxiliary valve to be governed by a spring that will deflect 0.01 inch for a vacuum in the carbureter of 1 inch of water.

(A) Assume that 230 cubic inches of air per second are passing through this carbureter at a velocity of 90 feet per second. By equation (8) the fuel velocity will be

$$Vf = \sqrt{(.00169 \times 90^2) - 1.9} = 3.43 \text{ feet per second,}$$

the vacuum will be

$$\frac{90^2}{64.4 \times 68.28} = 1.84 \text{ inches of water.}$$

The deflection of the valve will be

$$0.01 \times 1.84 = 0.0184 \text{ inches.}$$

The total admission area will be

$$\frac{230}{90 \times 12} = 0.213 \text{ square inch.}$$

The auxiliary area will be

$$0.213 - 0.077 = 0.136 \text{ square inch.}$$

(B) Assume now that ten times the original quantity of air is demanded.

The quantity of air would be

$$230 \times 10 = 2,300 \text{ cubic inches per second.}$$

This air must pass the fuel jet with a velocity sufficient to induce a flow ten times the initial quantity of the fuel.

As, by equation (12)

$$\dot{V}a = \sqrt{591.71\,(Vf^2 + 2gh')}$$

the air velocity that will increase the fuel flow ten times may be expressed

$$Va_{10} = \sqrt{591.71\,\{10\,(Vf^2) + 2gh'\}}$$

Substituting the values of the present example

$$Va_{10} = \sqrt{591.71\,\{10\,(3.4^2) + 1.9\}} = 263.67 \text{ feet per second.}$$

The vacuum will be $\dfrac{264^2}{64.4 \times 62.28} = 15.85$ inches of water.

The deflection of the valve

$$15.85 \times 0.01 = 0.158 \text{ inch.}$$

The total admission area

$$\frac{2300}{264 \times 12} = 0.73 \text{ square inch.}$$

The auxiliary area

$$0.73 - 0.077 = 0.653 \text{ square inch.}$$

Working Equations

As a practical convenience these equations may be simplified and expressed in terms of fuel velocity as follows:

$$\text{Velocity of the air} = 24.32\sqrt{Vf^2 + 2gh'} \qquad (13)$$

$$\text{Total admission area} = \frac{Qa}{292\sqrt{Vf^2 + 2gh'}} \qquad (14)$$

$$\text{Vacuum in inches of water} = \frac{Vf^2 + 2gh'}{7.44} \qquad (15)$$

$$\text{Total spring deflection} = \frac{(Vf^2 + 2gh')\,d}{7.44} \qquad (16)$$

where d = the spring deflection for a vacuum of 1 inch of water.

By the use of these formulæ the auxiliary air admission area may be determined for any number of points in the travel of the valve and the walls surrounding the valve may be made to conform to the curve so plotted, thus assuring the permanent maintenance of any desired air/gas ratio without adjustments of any kind.

Relation of Velocity to Vacuum

Friction

In all the foregoing calculations the influence of friction and other factors modifying the flow of liquids in a carbureter have been omitted for the purpose of permitting simplified statements of fundamental principles. These modifications are, however, of prime importance, none the less so because their variant values are undetermined. They affect the flow of both fuel and air to such an extent that, without giving them due consideration, the application of any formulæ expressing the relationship of actual flow of fuel and air would be impossible.

Instrumental Elimination of Unknown Quantities.—Barometric Effects.—Inherent Accuracy

Thus the formulæ herein expressed have, so far, tentatively assumed that the drop in pressure or "vacuum" at the mouth

of the fuel nozzle was the same as that within the mixing chamber. Repeated experiments have demonstrated the fallacy of such an assumption, to which indeed must be attributed the failure of many otherwise meritorious devices. Solution of the intricate problems existing between the mouth of the fuel nozzle and the mixing chamber, involving marked physical changes in both the liquid fuel and the air, would be interesting theoretically, but, from a practical standpoint, we are fortunately able to eliminate the effect of these modifying influences instrumentally. This can be accomplished by two structural modifications. First, the control of the auxiliary area directly by the vacuum at the mouth of the fuel nozzle, which construction also presents the further practical advantage of rendering the action of the instrument practically insusceptible to barometric changes. Second, by a slight modification of the curve of auxiliary-admission areas, so that the air velocities are increased to a sufficient amount, determined experimentally, to compensate for the frictional resistance offered by the nozzle to the flow of the fuel. Instruments constructed in accordance with the foregoing principles have been found to maintain a constancy of mixture in strict accord with the theory, and it has been determined that the slightest departure from the theoretical curve of admission areas produces negative results in constancy of composition.

Variable Mixtures

Modification of the Auxiliary Curve

If, however, it were desirable to vary the mixture composition for different operating conditions, the proposed method lends itself readily to that end. Thus, the auxiliary areas may be diminished at and near the starting end of the curve, resulting in the richer mixture so often claimed to be necessary for easy starting. At ordinary road speeds the areas may be so calculated that a mixture of high fuel economy will result, while at extreme open-throttle for high speed, contraction of the admission curve will increase the richness of the mixture for the development of maximum power. In other words, the designer

has but to determine the range of mixture composition which he considers most satisfactory and construct the admission curve in accordance therewith, knowing that whatever action has been selected will be repeated with invariable exactitude.

Constant Mixture

Advantages

The results obtained from many different engines by the use of gasoline mixtures of really constant composition have been so pronounced as to be in the nature of a revelation, particularly as regards certain details not ordinarily considered as primary functions of carburetion. There is noticeable a marked quietness of operation not easily explained, unless, possibly, the uniform rate of flame propagation establishes a rhythmical vibratory effect. The objectionable features of fluctuating mixtures are, naturally, minimized. After a full season's running the cylinders of several cars were found free from carbon, while the spark-plug points were clean and the porcelains discolored by heat only. Exhaust gas analysis shows practically no loss through incomplete combustion. The average of 44 samples taken from several different cars under all sorts of road conditions gave 0.43 per cent CO, while 29 samples yielded no CO.

Velocity the Only Constant

Governing the fuel flow by the *velocity* of the entering air seems to be an ideal method for constancy. At a given number of revolutions per minute a given engine invariably takes the charge into its cylinders at a definite velocity. Velocity, however, is the *only* fixed quantity. Chemical composition, pressure, temperature and, consequently, density of the charge, may vary widely, but whatever the nature of the charge—whether it be the rarefied atmosphere of the mountain-top or the dense fog of the seaboard—at a given engine speed the cylinder is filled (according to its volumetric efficiency under the conditions) in the same interval of time. Velocity is a constant, and upon it

and it alone may be safely based the computations necessary for accurate metering of the fuel.

Loss of Volumetric Efficiency

The practical operation of this type entails that increased air quantities be admitted at velocities sufficiently increased, so that the proper amount of fuel be inspirated.

Low Speed Velocities

To insure ease of starting, an initial velocity of the entering air of 90 feet per second is desirable, although with a properly designed manifold this may be safely reduced to 60 feet per second. This induces a fuel velocity as follows:

By equation (8)

$$Vf = \sqrt{(.00169 \times 60^2) - 1.9} = 2.0455 \text{ feet per second}$$

High Speed Velocities

Assuming the engine at maximum speed requires 15 times the initial quantity of air, and consequently 15 times the initial flow of fuel, necessitating a velocity of

$$15 \times 2.0455^2 = 62.76 \text{ feet per second.}$$

By equation (13)

Velocity of the air $= 24.32 \sqrt{67.76 + c + 1.9}$ where c is a coefficient of friction.

If, for illustration, we assign to c a value of .04 we have

$$Va = 24.32 \sqrt{65.2 + 1.9} = 199.32 \text{ feet per second.}$$

By Chart I, this velocity is seen to represent a volumetric loss of about 2 per cent. With an initial velocity of about 90 feet per second this loss is about 6.2 per cent.

Volumetric Loss

To these losses must be added the loss by friction in carbureter and manifold, and it is evident that practical design must

recognize these volumetric losses and select the range of velocities accordingly.

Summary of Types

Type I. *Non-Compensating.*—Produces an enriched mixture as speed increases.

Type II. *Mixing Valve.*—Produces a leaner mixture as speed increases.

Type III. *Vacuum Operated Auxiliary Valve*—Has an inherent tendency to impoverishment, which cannot be accurately corrected.

Type IV. *Multiple Fuel Jets.*—Each jet subject to the same conditions as Type I.

Type V. *Variable Fuel Orifice.*—May produce constant mixture but entails delicacy of adjustment.

Type VI. *Constant Vacuum.*—May produce constant mixture of one definite air/gas ratio, but entails mechanical difficulties if constancy is to be maintained where ratio is changed.

Type VII. *Compensating Nozzle.*—Compensation effected by dissimilar laws. Limited speed range on full throttle and inherent difficulties of acceleration.

Type VIII. *Compensation by Velocity.*—Constancy of mixture and freedom from barometric changes obtainable. Slight decrease of volumetric efficiency at high speeds unavoidable.

CHAPTER II

THE INTAKE MANIFOLD

THE functions of the intake manifold are so closely allied with those of the carbureter as to be inseparable in any detailed study of the science of carburetion.

Functions of the Carbureter and Manifold

With the fuels of the present day, the carbureter proper does little else than to proportion the amount of liquid fuel delivered to the air. Thus, it may be stated that the carbureter is responsible for the chemical composition of the mixture, while the physical condition of the charge is dependent upon subsequent processes of gasification and diffusion taking place very largely within the intake manifold, the valve chambers, and even within the cylinder itself.

The design of the intake manifold and its effect on the physical characteristics of the charge, therefore, become an essential part of the problem of carburetion.

PROBLEMS INVOLVED

Design of the intake manifold of the liquid engine presents two problems: First, that each cylinder receive an equal quantity of mixture; second, that the mixture reaching each cylinder shall possess the same chemical and physical characteristics. These factors are of much greater importance in the smoothness of operation and general efficiency of the engine than is commonly recognized.

THE CONDITIONS

Product of Carbureter, Mist—Not Gas

Were the product of the carbureter a homogeneous gas, the problem of manifold design would be largely a matter of con-

venient dimensions and proportionate branchings. In fact, when the grade of commercial gasoline was much lighter than it is, the manifold presented few problems. The prevailing grade of gasoline and its constant degeneration, coupled with the commercial necessity of using fuels of still lower volatility, make the manifold an active and important adjunct of the carbureter. The mixture leaving the throat of the carbureter is by no means a true gas, but consists chiefly of liquid particles carried in mechanical suspension in the moving air current. From the moment of admixture, these particles undergo constant evaporation. With the highly volatile fuels formerly obtainable, the reduced velocities through an enlarged area in the mixing chamber of the carbureter afforded sufficient time to convert these particles almost, if not wholly, to gas. With the less volatile fuels of to-day the time factor of unaided evaporation is so great that a considerable portion of the fuel traverses the greater part, if not the entire length, of the manifold as a mist suspended in the air current.

Velocities in the Manifold

Necessities for Proper Velocities

A definite velocity is required to maintain this suspension, dependent upon the size of the liquid particles, which, in turn, depends upon the atomizing force to which the fuel has been subjected. The moment the speed of the moving air current is decreased below this critical velocity, the larger particles are deposited and the mixture no longer contains the proportion of fuel that was so carefully metered into it by the carbureter.

Any enlargement of the cross-sectional area traversed by the mixture decreases its velocity, and hence, if, as in starting, the fuel mist is to be carried to the cylinders as such, the diameter of the manifold would be confined to narrow limits.

A liquid fuel engine of average flexibility requires at least from twelve to fifteen times its minimum amount of air at maximum speed. If a velocity of, say, 30 feet per second is necessary to maintain the suspension of fuel atomized to a given fineness,

and if the area of the manifold is such that this velocity is to be maintained at the lowest speed, then at maximum speed the velocity would approach 450 feet per second. This would entail

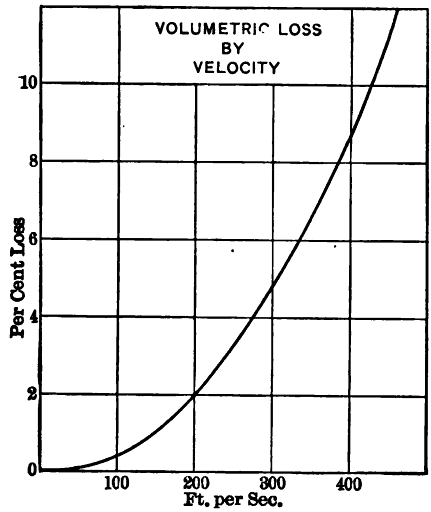


CHART I.

a loss of volumetric efficiency from velocity head alone of over 11 per cent, without considering friction which would increase this loss.

Deposition of Liquid Fuel

Deposition Not Condensation

Furthermore, whenever a moving fluid touches foreign surfaces the velocity of the surface stratum is markedly diminished by the friction of the contact, called "skin-friction," an amount dependent upon the condition of smoothness of the frictional surface. Hence, that portion of the air column which touches the walls of the manifold frequently falls below the critical velocity, even though the interior of the column may be maintained well above it. The result is the well-known wetting of all surfaces, commonly, but erroneously, attributed to condensation. Condensation implies a change of state from

a gas to a liquid. As the fuel has never been a gas during the process under consideration, the term condensation is not only clearly a misnomer, but misleading as to actual conditions and causes.

From the foregoing it is evident that it is wholly impractical to depend upon high velocities within the manifold for either the quantitative or the qualitative maintenance of the mixture. We recognize, then, that there is and *must* be a deposition of liquid upon all interior surfaces, depending in amount upon

(a) The degree of atomization within the carbureter.

(b) The cross-sectional area of the manifold.

(c) The form and condition of smoothness of the manifold passages.

Evaporation of Deposited Fuel

Before attempting design, let us further consider what takes place within the manifold. The surface of the liquid film wetting the walls is subjected to the attrition of the moving air-column, with resulting evaporation of the liquid. This evaporation takes place only from the surface of the liquid and is a relatively slow process with low grades of fuel. It is clearly desirable, therefore, to avoid pockets where any depth of liquid can accumulate, but, instead, to increase the available surface to the greatest possible extent, and hence, we hear of the advisability of roughened interior walls.

Surging

The alternating processes of deposition and evaporation are evidenced in the "surging" with which we are familiar when starting some engines on a cold morning. After running a short time, the rate of evaporation, assisted by the elevation of temperature beneath the hood, equalizes with the rate of deposition and the engine assumes a more even tenor of operation.

Effect of Bends

Another factor increasing the difficulty of proper distribution of a mist-laden mixture is the tendency of the liquid to seek

the outer periphery of all curves. However finely the liquid may be comminuted, so long as it remains a liquid, its specific gravity is far greater than that of the air, and, being thrown violently against the outside of the curve by centrifugal force, its velocity is so lessened by the impact that there is a greater tendency to impoverish the mixture than would be the case with a straight pipe, or can be accounted for by the additional resistance of the curve. It follows that the shorter the radius of the curve the greater the tendency to cause deposition.

Causes of Hard Starting

With the frequent enlargement of area and the tortuous passages of many manifolds, it is probable that all the unevaporated liquid is deposited before reaching the cylinders. Hence, an engine so equipped is hard to start when cold. It is a common experience with oversized and otherwise poorly designed manifolds to observe an actual dripping from around the throttle shaft and from the primary inlet to the carbureter, after one has become exhausted by ineffectual cranking of a cold engine. Is it any wonder that a starter frequently refuses duty?

Metering and Carburetion

Carburetion Within the Manifold

Under these conditions, as has been noted, the carbureter functions chiefly as a metering device, while true carburetion of the air by fuel vapor really takes place largely within the manifold.

To some considerable extent this process is one of surface carburetion. The surface carbureter was abandoned early in the art. Its faults are too well known to need further discussion at this time, and reversion to the functions of this abandoned device, which has been unconsciously thrust upon us by the present low grade of fuel, is a curious coincidence. Such conditions seem unavoidable, however, and must be frankly met.

Heating

For several years manufacturers have provided means for heating the carbureter by circulating water, while the latest kerosene carbureters employ the higher temperatures of the exhaust in a jacket surrounding the air-passages.

Application of Heat

In view of the fact, as determined herein, that a very considerable part of the carburetion actually takes place after the mixture has left the carbureter, it is difficult to see why more manufacturers have not followed the few examples already set them and employed means for heating the manifold. Heat so applied is most effectually communicated directly to the deposited liquid film, hastening evaporation and insuring the rapid diffusion of the fuel vapor with the entraining air in a manner that leaves little to be desired.

Starting, Cold

Conditions for Easy Starting

Coming now to the question of practical manifold design, we are at once confronted by the starting period wherein no heat is available. Without the aid of heat the only practical method of securing comparative ease of starting is to so design the manifold that the greatest amount of fuel mist may be delivered to the cylinders. As we have seen, this entails:

(a) Powerful atomization in the carbureter, because the smaller particles are more easily entrained at low velocities.

(b) The least possible manifold diameter consistent with volumetric efficiency at subsequent high speeds, insuring a more thorough entrainment of the fuel mist.

(c) Smooth interior surfaces, reducing skin friction and absence of enlargements of cross-sectional area, as maintaining the velocity already acquired.

To these must be added:

(d) Minimum length of manifold passage.

In any event, the best that can be hoped for in starting cold is that a small portion of the fuel will reach the cylinders, either in a gaseous or liquid form, sufficient in quantity to start the cycle. Hence, the utility of the excess of fuel secured by "priming" and, incidentally, the necessity of having this priming charge highly atomized.

Continuous Operation

Let us now consider the efficiency of such general design in delivering to the cylinders a truly gaseous mixture after the engine is heated.

Atomization

(a) That fine atomization is a necessary prerequisite is evident when we consider that the fuel particles are spherical in shape. The volume or weight of a sphere decreases with the cube of its diameter, while the surface exposed to evaporative influences decreases only as the square of the diameter. The rapid increase of effective surface exposure, as diameters are decreased, is apparent.

Area

(b) The proper diameter of the manifold is a question for the individual judgment of the designer. The permissible loss of volumetric efficiency, due to velocity head and friction within the manifold, should be adjusted to other factors of volumetric loss, such as valve location, areas, and timing. The total loss should be so established that the highest possible velocities can be tolerated within the manifold.

Condition of Smoothness

(c) As to the choice between smooth and roughened interior walls, the writer believes, from his experience, that with proper heat distribution during continued operation there is little danger of unevaporated fuel reaching the cylinders with the smoothest of interior walls. The numerous bends, unavoidable in multi-cylinder construction, and even the frictional opposition of the conventional butterfly throttle, will insure deposition of

that portion of the fuel which has escaped previous evaporation, and, as has been noted, the application of heat to the surfaces which receive this deposit will promote its thorough evaporation.

Length

(d) Consideration of the actual distance between the carbureter and the valve-chambers shows a possibility of real danger in making the manifold too short. It is conceivable that if the foregoing conditions are complied with, the manifold might be made so short that unevaporated liquid would actually reach the cylinders, resulting in inefficient combustion.

Qualitative Distribution

Furthermore, we have already noted that the mixture entering the manifold is far from homogeneous. To produce the homogeneity necessary for equal qualitative distribution, we must provide conditions favoring the rapid diffusion of the air and fuel vapor.

Diffusion

Just how rapid this diffusion must be is best illustrated by considering the time element of the passage of gas through the manifold. For example, assume that the length of the manifold passage is two feet. At a minimum velocity of 1,800 feet per minute a given unit of gas remains in the manifold but 0.066 of a second. At the not uncommon velocity of 8,000 feet per minute (which only entails a loss of volumetric efficiency of less than 1 per cent), a unit of gas remains in the manifold but .015 of a second.

Economizers

Under these conditions a most intimate mixture of the gases is necessary, and hence, the real efficiency of some of the so-called "economizers" on the market. The offset, or reverse bends, in the upright member of some manifolds, is usually merely for convenience in locating the carbureter in the limited space available. The bends so introduced, if properly designed, are not a detriment, as is frequently stated, but, instead, possess the

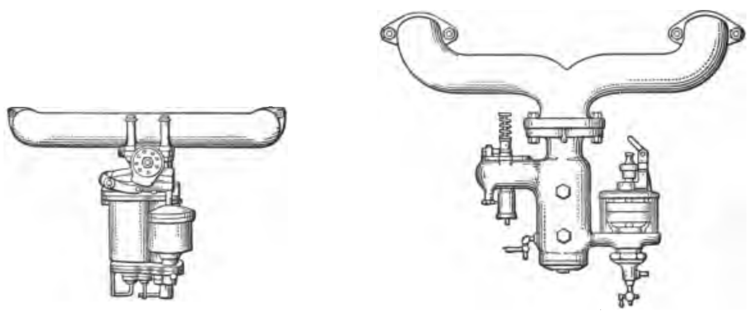
distinct advantage of promoting diffusion through a more thorough mixing of the gases.

Qualitative Distribution

It has been the writer's experience that many faults of operation were due solely to uneven qualitative distribution of the mixture. This fault, infrequently recognized, results in a wide range of troubles from poor economy or a slight lack of power to persistent and perplexing missing. This being the fact, the practice of locating the carbureter immediately at the branchings of the manifold cannot be recommended.

Types of Manifolds

The types of manifold shown in Figs. 1 and 2 embrace this objectionable feature. Similar designs are becoming more

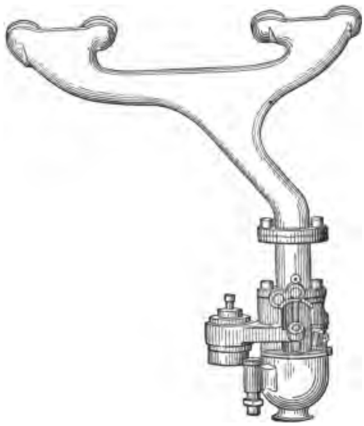
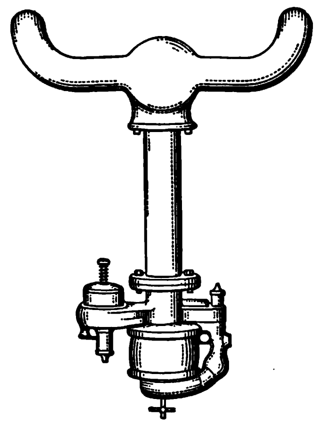


Figs. 1 and 2.

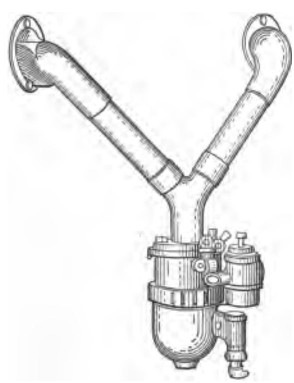
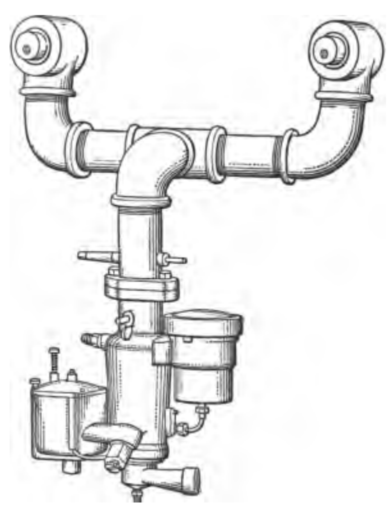
common with the adoption of pressure feed on the fuel. Engines so equipped are notably easy starting, but the writer believes operative efficiency is sacrificed as a result.

Fig. 3 shows the opposite extreme in an attempt to provide diffusion chambers. With highly volatile fuel, or with proper heating of the vertical member, these chambers would doubtless afford distinct advantages through the mixing of the gases by expansion and contraction. For cold weather starting, with the fuel of the present day, the writer has daily reason to criticize this design.

A modification of the diffusion-chamber idea is shown in Fig. 4. If constructed with the proper dimensions and with the vertical member of this manifold heated, qualitative dis-

FIGS. 3 AND 4.

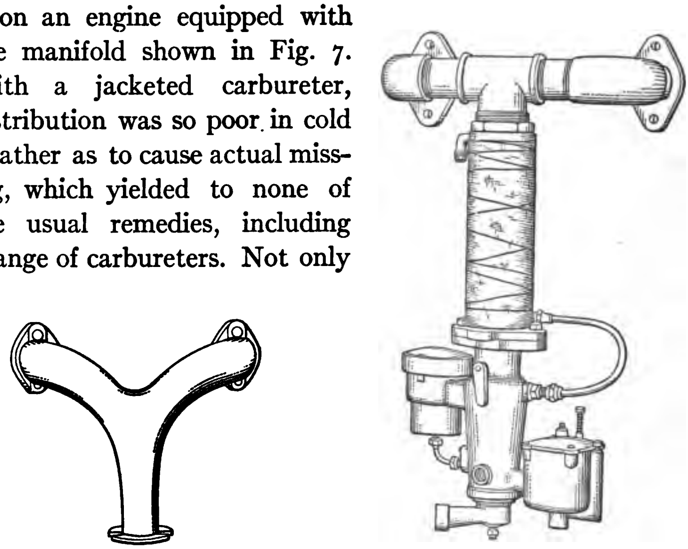



FIGS. 5 AND 6.

tribution between the two branches could hardly fail to be excellent. Starting cold, however, would be something of a problem.

Fig. 5 shows a type exhibiting noticeably erratic distribution when used with the unjacketed carbureter. With this manifold of brass, with a smooth interior finish, the engine started easily but developed a noticeable lack of power, particularly at low speeds. An experimental manifold shown in Fig. 6 was constructed of ordinary 1¼-inch pipe fittings, being practically the same size as the original manifold. In this crude affair diffusion was secured by the additional length of and bends in the central member, and also in the slight enlargement of the central tee. With a highly atomizing carbureter the smoothness of operation and gain in power were most marked. Owing to the difference in the carbureters employed, this test is of lessened value so far as the manifold itself is concerned. It is of value because of the close and indissoluble relationship existing between the work of the carbureter and the functions of the manifold.

More definitely conclusive was the experiment performed upon an engine equipped with the manifold shown in Fig. 7. With a jacketed carbureter, distribution was so poor in cold weather as to cause actual missing, which yielded to none of the usual remedies, including change of carbureters. Not only



FIGS. 7 AND 8.

was this trouble completely obviated, but marked increase in power and better general all-around action was obtained by no other change than surrounding the vertical member of this

manifold with a close-wound coil of five-sixteenth copper tubing carrying hot water from the circulation.

The improvement in operation was so marked that the experimental manifold shown in Fig. 8 was constructed with a more effective water-jacketing. Owing to its experimental construction of brass pipe and standard fittings, it was impossible to maintain the downward slope of the branches, but notwithstanding this, the owner preferred to continue the use of the makeshift rather than the original manifold. Of course, the short radii of the tee and the elbows were indefensible, but, while starting cold, though not at all bad, might have been improved by a permanent design, distribution was all that could be desired.

Quantitative Distribution

From the foregoing it is seen that the conditions required for easy starting do not, for the most part, conflict with the requirements for continued running. There remain to be considered details of design necessary to secure the same quantity of mixture in each cylinder. Having made provisions to insure a homogeneous and truly gaseous mixture, the remaining questions simplify themselves largely to problems of equal frictional resistances in the different branchings.

Resistance of Bends

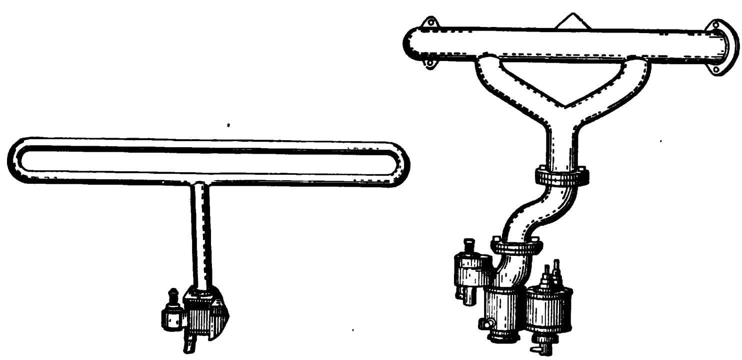
Resistance to the flow of air through pipes may be readily determined from the formulæ and tables given in the standard text-books. In computing this resistance, due attention must be given to the additional resistance offered by bends. Kent, 8th edition, page 593, gives a convenient table on the effect of bends, wherein lengths of straight pipe, equivalent in resistance to bends of different radii, are given. As an illustration of the use of this table we note that the resistance of a standard 1¼-inch pipe elbow (mean radius 1 1/32 inch) is equivalent to a little more than 4 feet of straight pipe. If the mean radius were increased to 4 9/64 inches, the resistance would be reduced to that of 11 3/8 inches of straight

pipe. By this method the total resistance of the branches may be determined and equalized.

It must be borne in mind, however, that bends are prolific of deposition of entrained liquid, and therefore the drainage of these bends should be carefully directed toward the heated surfaces. In furtherance of this idea, the branches should be given a drainage slope away from the cylinders. Fig. 3 shows that careful consideration has been given to these details. Note the longre radius of the bend of greater angularity and the location of the junction of the upright member to the right of the center. Note also the downward slope of the lower surfaces of the cross members toward the upright member. All these details tend to equalize distribution, both quantitative and qualitative.

Six-Cylinder Distribution

Six-cylinder engines present greater complications in the matter of quantitative distribution than do the fours. In fact, the development of the early sixes was retarded by a lack



Figs. 9 and 10.

of understanding of actual conditions within the manifold. The greater distances to be travelled by the gases, the more numerous branchings and the overlapping of the suction strokes, all emphasize the tendencies toward uneven distribution.

Fig. 9 shows one of the methods employed to obviate this

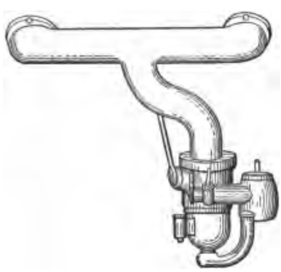
difficulty. In this manifold it will be noted that the supply to each cylinder is drawn from *both* branches. As the resistance is increased by the greater distance travelled in one branch, it is proportionately decreased by the shorter distance travelled in the other branch, and hence is constant.

Another arrangement giving the same effect is a horizontal pipe carrying a longitudinal partition, or baffle plate, as an integral part of the casting.

The same result is sought by a different arrangement, shown in Fig. 10.

Conclusions

The conditions outlined in this chapter are fundamental. They can be met in a variety of ways which will suggest them-



Figs. 11 and 12.

selves to the designer. Briefly summarized these conditions consist of:

1. A heated manifold.
2. High manifold velocities.
3. Smooth interior walls.
4. Long radius curves.
5. No enlargement of cross-sectional area.
6. Absence of liquid retaining pockets.
7. Branches sloping toward the central member.
8. Adequate provision for mixing the gases.
9. Equal resistance in the branches.

The writer has been unable to secure an illustration of a manifold embracing all these features if, indeed, such exists.

The general idea is expressed in Fig. 11, which, in point of fact, is a manifold of one of the best known cup-winning cars. Improvement might be made in this design by a slight drainage slope given to the branches and by water-jacketing the central member. It is, of course, assumed that the diameter of this manifold is properly proportioned to the displacement of the engine.

Fig. 12 embraces *every apparent fault* that can be introduced into a manifold. Its diameter is great. Its bends are sharp. Drainage is directly away from the central member. Pockets are formed at the base of the branches. It has no provision for diffusion, is unheated, and, if in consonance with the rest of the design, its interior walls are doubtless rough.

CHAPTER III

CARBURETER TESTING

On the Block

THE usual test to which a carbureter is subjected while attached to an engine on the block consists of:

(1) A series of readings of maximum horse-power at various speeds. With the throttle wide open, various loads are imposed upon the engine and the resulting horse-power curve plotted therefrom. The fuel consumption is also noted at each speed and the resulting curve plotted.

(2) This programme is sometimes elaborated by a series of runs at three-fourths, one-half, and one-quarter throttle with the results given expressing horse-power developed and the fuel used.

(3) More infrequently, the rate of acceleration is noted as the number of seconds required to reach a given speed, either running light or with some empirical load.

(4) Very rarely flexibility is determined by a mechanical device which slowly closes the throttle and then suddenly opens it, and then reverses its operation by opening the throttle slowly and snapping it shut.

Maximum Horse-Power

Determination of the maximum horse-power curve (1) is, of course, an essential detail of any carbureter test. It shows any erratic behavior in the functioning of the instrument and detects any undue internal resistance.

For motor-boat requirements, where maximum horse-power at full speed is of primary importance, this test gives the most desired information.

In automobile practice, however, it is a rare occurrence that an engine is called upon to deliver its maximum power at its highest speed, except in the case of racing cars. Nine-tenths of all driving is done with the throttle partially closed, and

consequently the object of the test on the block should be to determine the relative performance of carbureters under various throttle openings.

A comparison of carbureter tests conducted for maximum horse-power alone will disclose surprisingly little difference either in power developed or in fuel consumption. The same carbureters will, however, show markedly different results under road conditions.

In chapters I and II the causes for these different performances have been analyzed. To actually determine the relative merits of various devices on the block, it is necessary to simulate road conditions in so far as it is possible to do so. The first of these conditions to be observed is that exhaustive tests must be conducted at different throttle openings.

Fallacy of Set Throttle Tests

If comparisons are to be accurately made, the plan usually followed as outlined in (2) is fallacious, because, owing to lack of any standardization of throttle sizes, shapes, or even types, the same position of the throttle arm or crank does not necessarily mean equal, or even approximately equal, openings on any two instruments. Nor is a car driven with any reference to, indeed seldom with knowledge of, the amount of throttle opening afforded by intermediate positions of the throttle lever on the steering-wheel. Instead, the throttle is opened until a certain result is accomplished, *i.e.*, the moving of a given load at a given speed.

Testing With Fixed Load

To reproduce this condition on the block, a certain load should be set off on the dynamometer scale and the throttle opened until the beam balances at the desired speed. With the electric dynamometer the load increases automatically with the speed. This requires simultaneous adjustment of both rheostat and carbureter throttle. With the hydraulic dynamometer a somewhat similar condition exists, necessitating simultaneous regulation of hand-wheel on the brake and throttle

of the carbureter. These adjustments are, however, easily made after a little practice.

By repeating the foregoing with various loads carried throughout a range of speeds, points may be determined from which may be plotted a curve fairly representative of "Part Throttle Performance." This curve will prove of far more value in comparing the performance of different instruments than will the maximum horse-power curve alone.

In order to determine the true characteristics of the curve, the following procėdure is recommended. Having determined the maximum torque of the engine, this load is divided into equal parts—say fifths. Runs are then made throughout the entire speed range, with the motor carrying, say, one-fifth of its maximum load. Limits of speed, both slow and fast, are noted, together with the fuel consumption. This test is repeated for two-fifths, three-fifths, and four-fifths load.

The results obtained are frequently surprising. A carbureter that will show excellent economy on full throttle may fail utterly to carry a given load at a certain speed on part throttle, necessitating an enrichment of the mixture that will show the futility of the record established at full load.

The fuel consumption curves may be comparatively smooth at full throttle but widely variant while carrying constant load throughout the speed range. Speed limits, both high and low, will be found to vary greatly with different types of carbureters.

Under this method of testing, the different types discussed in Chapter I never fail to exhibit all the peculiarities enumerated therein.

Acceleration

In addition to the foregoing, acceleration (3) should be determined as outlined, but this determination should be made, when possible, with all the different loads mentioned in the preceding test. In this connection, however, it is well to note that the automatically increasing load of the electric or hydraulic dynamometer is unobjectionable for the purpose of determining acceleration, as it closely simulates road conditions where the

load increases by wind resistance approximately as the square of the speed.

Flexibility

The test outlined in (4) is of great practical merit for purposes of comparison, if properly conducted. If the variable throttle moving device is mechanically operated so its movements may be continued over a considerable time-period, it is frequently found that after several repetitions some carbureters will choke, even though this tendency may not be in evidence during one or two trials. This may be due to undue enlargement of area and consequent reduction of velocity. Whatever its cause, it is a prolific source of annoyance when driving a car through traffic, and should be detected by a properly conducted block test.

Like the preceding tests (1 to 3), test (4) should be conducted at various loads, for it will be found that, as in the other instances, performance will usually vary widely at different loads.

Again, if this flexibility test is of sufficiently long duration, the fuel consumption may be accurately measured. A determination of this kind gives a far more accurate measure of the *actual* performance of a carbureter in practical road use than is obtainable by any other system of averages.

In city use particularly, a car is rarely driven two consecutive minutes with the same throttle setting, and consequently, as is well known, fuel consumption is much higher than in the case of a cross-country run, which may be, in some measure, compared to a constant load in block testing.

With a full block test, conducted along the lines herein outlined, no function of the carbureter will escape scrutiny. Comparisons of different types will serve to establish their relative merits and their peculiar adaptability to the engine used in the test.

Practical Results

For automobile use, practical interest centres chiefly in part-throttle performance, acceleration, and flexibility. Economy is of course desirable, but becomes of primary importance only

in the larger units, while maximum power at maximum speed is a consideration confined wholly to racing cars.

In marine practice, the demand for maximum power from a given size of engine, coupled closely with minimum fuel consumption, is the chief consideration, followed closely by a demand for maximum speed. Part-throttle performance is of less importance, while flexibility and acceleration are the last consideration.

Automatic Apparatus

In the testing laboratory, measurements should be made as automatic as possible. A convenient method of accomplishing this result is to have the fuel tank balanced on a pair of scales. The beam of these scales in falling closes an electric circuit which starts a stop-watch, revolution counter, and bell. When the bell sounds, the operator reduces the weight on the scale beam by one pound (or whatever other unit seems desirable). When this unit is consumed, the beam falls again, closing the circuit. This disengages the revolution counter and stops the stop-watch, while the bell announces the end of the run.

Carbureter Testing on the Road

Block Testing Insufficient

No matter how comprehensive a block test may be, the practical man bases his final judgment of the merits of a carbureter by its actual performance on the car. This is wisdom born of experience. Though we may simulate certain road conditions on the block, there are certain factors encountered in road work that cannot be duplicated; for example, the load on the motor increasing with the square of the speed, the loss in transmission from motor to the rear wheels. Then tire losses, and even rolling resistance of the car itself varies not only with every make, but with every changing condition of roadway itself, or with the whims of the wind.

Difficulties in Comparing Results

Vibration, road shock, changes of float-tank level from gradients encountered, and sometimes sudden changes of temper-

ature, and finally dust and dirt, are among the road conditions which a carbureter must faithfully meet, and which cannot be reproduced in any test. Small wonder that experience has taught us to look askance at any test which necessarily omits these factors which must be met daily. At the same time,

FIG. 13. THE ACCELEROMETER.

when we scan the array of formidable conditions, we are less likely to place too much dependence in the *opinion* of any individual on the comparative road performance of competing devices. Two carbureters may be tried on the same car and for the same distance over the same road, but speeds cannot be maintained the same at every point.

Barometric and temperature changes cannot be accurately compared. Windage may not be the same during both tests.

The driver in each instance will not press his accelerator the same amount at the same place. In brief, conditions cannot even approximate constancy in both trials.

Comparative road testing can, therefore, be of value only when each test is conducted over a period sufficiently long to minimize the errors by the law of averages. This requirement is one not always easily fulfilled, and therefore it would seem desirable to find a method of car testing which will give an accurate comparison of performance of various devices, either actually on the road or under controllable conditions as closely approaching those of the road as is possible within the confines of a laboratory.

The Accelerometer

This instrument was designed by H. E. Wimperis, M.A., A.M.I.C.E., A.M.I.E.E., of England. The outward appearance of the instrument is shown in Fig. 13, while the construction is shown in Fig. 14.

The instrument has no mechanical connection with the car. It is simply carried in any convenient position on the car where it can be leveled by means of the adjusting screws on its base.

The dial of the instrument carries a double scale, reading each way from 0. The upper scale reads, "Acceleration in feet per second per second" on one side of 0, and "Retardation in lbs. per ton of 2,000 lbs." on the other side. The lower scales read "Upward Gradient," and "Downward Gradient."

Description

Principle of Operation

The instrument depends for its operation on the inertia of a copper weight *A* (Fig. 14). The centre of gravity of this weight is eccentric to its axis of revolution *B*. Any force in the direction of the arrow on the dial tends to make the mass of copper lag. This lag rotates the spindle *B*, which, by means of the gear train *C*, rotates the spindle *D*, winding up the hair-spring *E*.

The hair-spring is so calibrated that the pointer reads ac-

curately on the dial scales the actual forces affecting the copper disk.

Compensation

Any tendency of the disk to oscillate is checked by its passage between the poles of the permanent magnet *F*. The arrange-

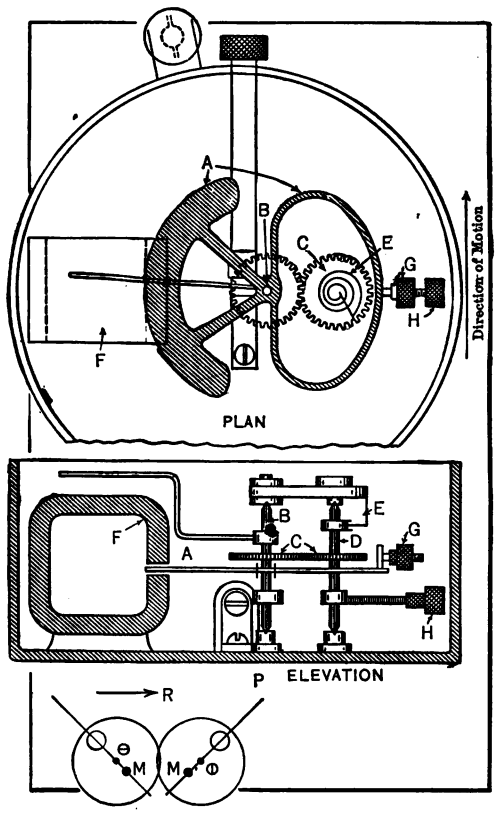


FIG. 14. ACCELEROMETER SECTION.

ment of the gearing effects what is called a "compensating balance" which neutralizes transverse forces and causes the instrument to record correctly even on heavily cambered roads. The reading of the instrument is in no way affected by grade, as will be seen by consideration of its principle of operation.

Let F = Force in pounds.

M = Mass or $\frac{W}{g}$

A = Acceleration in feet per second per second.

g = Acceleration due to gravity, 32.16 feet per second per second in middle latitudes.

W = Weight in pounds.

G = Gradient in percentage.

Then as

$$F = M A \tag{17}$$

$$A = \frac{32.2\,F}{W} \tag{18}$$

The force necessary to move a unit weight up a grade is

$$F = \frac{W\,G}{100} \tag{19}$$

Substituting the value for F in equation (18), the acceleration equivalent to this force is

$$A = \frac{32.2\,W\,G}{100\,W}$$

which reduces to

$$A = 0.322\,G \tag{20}$$

Assume that a car weighs 1 ton and is equipped with an engine that will produce an accelerative force of 2 feet per second per second on a level.

Acceleration. Up-Grade

Suppose now that this car is ascending a 2 per cent upward gradient, which graduation is coincident with 0.624 in the acceleration scale. [See equation (20).] The engine is therefore exerting a force equivalent to an acceleration of 0.624 ft./sec./sec. in maintaining constant speed. When the throttle is wide opened the speed of the car will increase and the needle will stand at 2 ft./sec./sec. That is, the acceleration *from* the initial speed and *on* the 2 per cent grade will be (2 − .624) =

1.376 ft./sec./sec., but the *total* accelerative power will be 2, as it was on the level.

Acceleration Down-Grade

Or assume this car to be descending a 3 per cent gradient. The force of gravity urging the car forward will be 0.966 ft./sec./sec. Hence, upon open throttle the total force moving the car forward will be $0.966 + 2 = 2.966$ ft./sec./sec., and the needle will swing through this arc, but the 0.966 being on the opposite side of 0, the needle will again stand at 2 ft./sec./sec. on the acceleration scale.

It is thus seen that acceleration can be measured, irrespective of grade, by suddenly opening the throttle wide and noting the reading on the acceleration scale.

Retardation is similarly read on the opposite scale and is subject to the same compensation as regards grade. Thus, by equation (19) the force acting in the opposite direction to the motion of the one-ton car on a 2 per cent gradient would be

$$F = \frac{2000 \times 2}{100} = 40 \text{ pounds per ton.}$$

Retardation Up-Grade

If the car were ascending a 2 per cent grade at constant speed and the power were suddenly shut off, the needle would return to 0, provided the car had no rolling resistance. As a matter of fact, the needle swings to the right of 0, an amount which consequently registers the resistance of the car in pounds per ton upon the upper or retardation scale.

Retardation Down-Grade

On the other hand, consider the car as being driven down a 2 per cent grade at constant speed, and the power suddenly discontinued. If the rolling resistance was greater than 40 pounds per ton (as it must be to necessitate the use of power), a preponderance of force would be exerted in a direction opposed to forward motion and the lag of the copper disk would cause the needle to move further to the right by an amount which,

minus the gradient reading, would be a true measure of the preponderance of retarding force, while the needle will give a direct reading of the total rolling resistance on the retardation scale, as in the previous instance.

A thorough understanding of these functions of the instrument makes its practical use easy.

Determination of Resistance

For the determination of rolling resistance, procedure is as follows: The accelerometer is placed on the car, with the arrow on the dial pointing in the direction of motion. It is carefully levelled, by means of the adjusting screw at its base, until the needle stands at 0, when the car is standing on the level. If the car-body is subject to much vibration, the instrument should be secured in this position by means of proper straps and its level position should be checked as frequently as possible.

Method of Reading

The car is next driven on the high gear at some constant speed, say 10 miles per hour, preferably down a slight grade. The clutch is suddenly thrown out of engagement and the reading taken on the retardation scale *before the car speed* changes. Owing to the momentum acquired by the copper disk, the first *swing* of the needle is to be disregarded, but it will be found to speedily settle on the true reading. A little practice will make this point readily determinable.

Total Resistance

A series of readings taken at various speeds will give the curve (R) of rolling resistance in pounds per ton, which may be readily reduced to total rolling resistance by multiplying by the total weight in tons.

Engine Friction

A similar curve (I) may be prepared by switching off the ignition instead of declutching. The difference between curves (I) and (R) will be the friction of the engine.

Transmission Friction

A third curve, N, is obtained by throwing the gear-shift lever into the neutral position. Curve R − curve N = friction

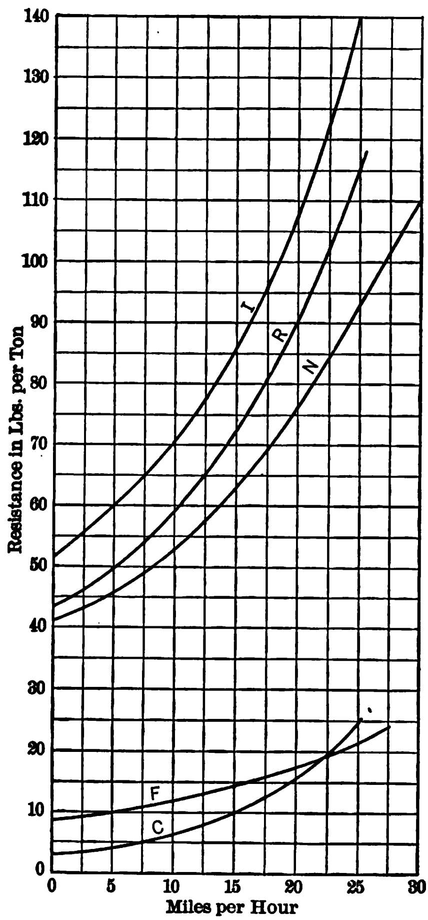


CHART II.

in the transmission, including the effect of the transmission brake, if any is used.

These curves are plotted in Chart II.

I is the resistance with ignition off.
R is the resistance declutched.
N is the resistance with gears in neutral.
F is the engine friction, or I-R.
C is the friction in transmission, or R-N.

Location of Mechanical Defects

These curves form the basis of all subsequent calculations. They are useful also in detecting and locating mechanical defects in the mechanism of the car. The curve R may be determined with either set of gears in mesh, and the friction of each be thus determined. By this method the cause of a decrease of power may be located, in worn bearings or gears, sprung shafts, insufficient lubrication, or dragging brake bands.

Determination of Acceleration

Having found the rolling resistance of the car, the next step is to determine acceleration. This is done by driving the car at a given speed and suddenly opening the throttle wide. Acceleration is then read directly from the acceleration scale. As in the case of retardation, acceleration readings are taken throughout as wide a speed range as possible and a curve plotted.

By equation (18), $A = \frac{32.2\,F}{W}$, therefore

$$F = \frac{A\,W}{32.2} \tag{21}$$

Hence, the force exerted by an engine giving an acceleration of A to a car weighing one ton will be

$$F_1 = \frac{2000\,A}{32.2} = 62.2\,A \tag{22}$$

Draw-Bar Pull

This is the force over and above that necessary to overcome the rolling resistance R. The draw-bar pull P per ton is therefore

$$P = 62.2A + R \tag{23}$$

Brake Horse-Power

The brake horse-power exerted at the clutch may be determined as follows:

$$\text{Miles per hour} = \frac{5{,}280}{60} = 88 \text{ feet per minute}$$

and

$$BHP = \frac{88\,PSW}{33{,}000} \text{ or}$$

$$BHP = \frac{PSW}{375}. \qquad (24)$$

where

BHP = Brake horse-power.
S = Speed in miles per hour.
W = Weight of car in tons.

Indicated Horse-Power

The indicated horse-power (IHP) may be found by substituting the values of points on curve I (ignition off) for those of R in equation (23).

Chart III shows the resistance curves I and R; the acceleration; the draw-bar pull; and the indicated and brake horse-power of a car weighing 1.57 tons, equipped with a four-cylinder engine, 4 x 4½ inches; gear ratio, 3.5 on direct drive. Wheels, 33 inches diameter.

Bases for Comparative Performances

Brake Mean Effective Pressure

A mathematical basis for comparing engine performances is afforded by determining the "brake mean effective power." Let

$$\eta = \text{Mechanical efficiency} = \frac{BHP}{IHP}$$

p = Indicated mean effective pressure in lbs./sq. inch,

then ηp = Brake mean effective pressure in lbs./sq. inch;
now if g = gear ratio,
D = Displacement of the cylinders in cubic inches,

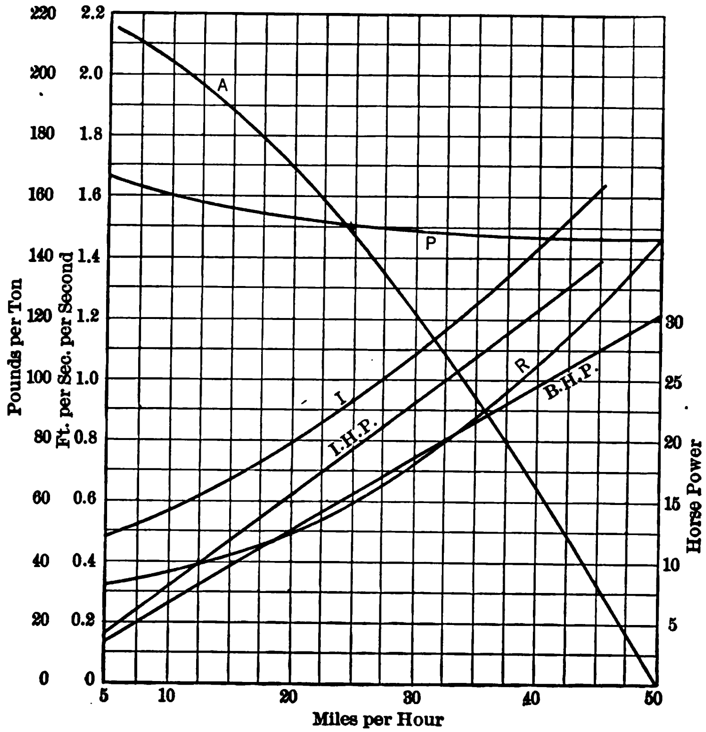


CHART III.

d = diameter of drive wheels in inches,
P = *Total* draw-bar pull in pounds,

then
$$\frac{\eta p \times g \times D}{2 \pi d} = P$$

and consequently
$$\frac{P\, 2 \pi d}{g D} = \eta p \qquad (25)$$

Thermal Efficiency

The accelerometer furnishes a means for directly determining the thermal efficiency of the engine by means of the resistance

curve (R) coupled with the actual mileage obtained on a known quantity of gasoline.

The foot-pounds of work performed are

$$R \times 5280 \times M$$

when

R = Average resistance of the run.
M = Miles per gallon of gasoline.
s = Specific gravity of the gasoline.
H = Its heat value in B.T.U. per pound.

Then the thermal efficiency of the engine will be

$$\frac{R \times M \times 5{,}280}{8.3455 \times 778\, H}$$

which reduces to

$$.81326\, \frac{R\, M}{s\, H} \tag{26}$$

A more convenient formula of sufficient accuracy for most tests is

$$\text{Thermal efficiency} = \frac{R\, M}{18{,}000} \tag{27}$$

This formula assumes gasoline of 0.72 specific gravity and a heat value of about 20,500 B.T.U. per pound.

General Observations

Accuracy

Practice with the accelerometer will lead to a surprising degree of accuracy in the results obtained. Two separate observers with different instruments have obtained results from the same car varying less than 5 per cent.

It is, however, necessary to accept only the mean of many readings. The instrument is not as sensitive as would seem desirable, and apparently might be equipped with jeweled bearings to advantage. It should also be provided with means for holding it securely to the car-body.

Levelling

Especially should great care be exercised in its initial levelling. This necessitates the selection of a perfectly level spot. The needle should then be swung each way from o until it invariably comes to rest on the o mark. This adjustment should be repeated as often during the test as conditions will allow.

Effect of Wind

It is, of course, apparent that the force and direction of the wind will materially affect the results obtained, hence it is desirable to select either a still day, or a road at right angles to the direction of the wind.

These practical difficulties have made necessary some method of reproducing road conditions in the laboratory. Such a method was proposed by the author and Prof. E. H. Lockwood, of the Sheffield Scientific School of Yale University. The following is from a paper prepared by them for the Society of Automobile Engineers. The method furnishes such an excellent means for carbureter testing that it is quoted here complete.

CHAPTER IV

THE PRACTICAL TESTING OF MOTOR-VEHICLES

MOTOR-CAR testing should be conducted for two purposes:

First.—To determine the actual performance of the car as a whole.

Second.—To determine the relative merits of the different components of the car.

The first is of practical interest to the sales department, the owner, and the general public. Interest in the second is confined largely to the department of engineering. But from the engineer's standpoint, much useless experimenting could be avoided by an accurate knowledge of the relative performance of different motor-vehicles, as at present designed, before attempting any comparison of constructional details.

ROAD TESTING

Any attempt at determining the actual performance of a car on the road is confronted with the problem of the uncontrollable variables introduced. Chief among these are the following:

(a) Condition of roadway.

(b) Force and direction of the wind.

(c) Frequent and uncertain change of gradients.

Among the instrumental difficulties encountered are:

(d) Lack of accurate apparatus for the determination of power, without a specially constructed car.

(e) Inability to measure fuel consumption accurately, owing to the vibration of the car-body.

PROPOSED TESTS OF PERFORMANCE

The purpose of this paper is to suggest a method of testing actual car performance on the block with results that can be reproduced on the road.

The method has been developed, and is at present employed in the Mason Laboratory of Mechanical Engineering of the Sheffield Scientific School, Yale University.

FIG. 15. TAKING ROLLING RESISTANCE.

FIG. 16. CAR ON TEST STAND.

The testing apparatus is located on the ground level near the Temple Street entrance to the laboratory. An open stretch of

level granolithic concrete floor, about 75 feet long, permits of towing tests to determine rolling resistance of the car at low speeds. For power tests the car is placed on traction drums where appliances are at hand to measure power and pull at different speeds. The general appearance of the car undergoing various tests is shown in Figs. 15 and 16.

ROLLING RESISTANCE

The first test is to determine the force required to pull the car slowly on the smooth level floor of the laboratory. This is accomplished by a recording dynamometer attached to the front of the car, as shown in Fig. 15. An enlarged view of the dyna-

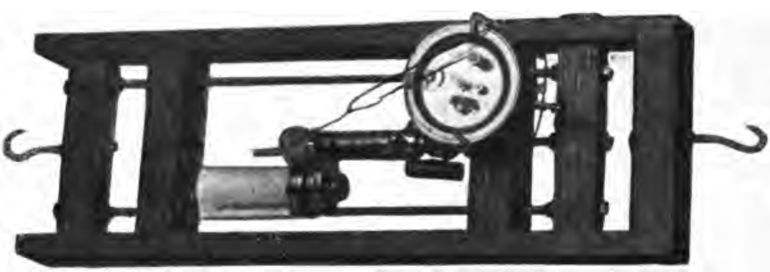


FIG. 17. THE DYNAMOMETER.

mometer is given in Fig. 17. The recording elements consist of a Tabor gas-engine indicator, held by a suitable frame so that the pull compresses the spring, marking a line on the drum

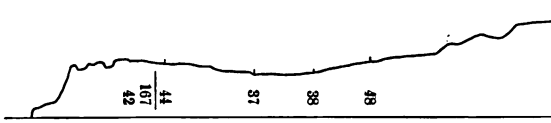


CHART IV.
Dynamometer Diagram.

while the latter rotates under control of a clock. A sample diagram from the dynamometer is shown in Chart IV.

Traction Drums

For power measurements the rear wheels are placed on drums whose top faces are level with the floor, while the front wheels remain at rest holding that end of the car in place. Connections are made from the rear axle to a permanent anchorage by chains

Fig. 18. Mounting of Rolls and Brake.

and turnbuckles, affording adjustment to centre the wheels on the drums and to resist forward movement when power is applied. The drums have faces 15 inches wide, treads centred 53 inches apart, and the actual circumference of the drums is 17.51 feet; 301 revolutions of the drums are equal to 1 mile. It was originally planned to measure the draw-bar pull directly from the axle connections, but this has never been carried out owing to practical difficulties.

Power measurements are made on a Prony brake-pulley, 36 inches diameter by 8 inches face, with a water-cooled rim, encircled by a rope brake. The brake is conveniently adjusted from the operating-table on the main floor by a hand-wheel and shaft telescoping over a worm-shaft on the brake-arm. The pull

of the brake-arm is registered on platform scales beside the operating-table. The arrangement of levers gives 123.4 pounds pull on the brake-arm for 100 pounds on the scales. The arm of the brake is made exactly equal to the radius of the traction drums, so that the brake-load is the same as the draw-bar pull. The brake and traction drums are shown in Fig. 22. The strap was originally made of steel band lined with maple blocks, as shown in the illustration. This has since been changed to a rope band of four parallel strands of ¾-inch rope suitably tied together. The action of the rope has been smoother, and leaves little room for improvement.

Drum Friction

The force required to rotate the traction drums with the brake-strap removed is a necessary quantity. This has been determined approximately by placing a car exactly central on the drums and measuring the draw-bar pull at different speeds by a spring balance. Thus far the friction force has been taken as 35 pounds, this being the average for cars of different weight, the change due to windage at various speeds having been too uncertain to be allowed for.

Draw-Bar Pull

The brake-arm, being equal to the radius of the traction drums, permits the direct determination of the draw-bar pull from the brake-load when the axle friction of the drums is included. The draw-bar pull can be computed from this expression:

Draw-bar pull = 1.234 × load on scales + 35 pounds.

The load on the scales can be read directly, using tare for dead weight of the brake-arm. The only uncertainty consists of the allowance for friction and windage of the drums. This element is, however, a small part of the total draw-bar pull, except at very light loads, and the figures given above are nearly correct.

Measurement of Speed

A Hopkins electric tachometer measures the speed of the traction drums, with the indicating dial mounted on the gauge board in front of the brake-operator. This reads revolutions per minute of the traction drums correctly within 3 per cent at all speeds. Accurate measurements of speed are made by a mechanical revolution counter, driven by linkage from the traction drums. This counter is located at the operating-table beside the electric tachometer, where stop-watch observations are made simultaneously with the counter readings at the beginning and end of each run.

Gasoline Measurement

The fuel supply is contained in a five-gallon tank placed on scales weighing to sixteenths of an ounce; thence led by a rubber tube to the gasoline inlet of the carbureter. The rubber tube is sufficiently flexible to allow accurate weighing while it is attached to the can. An electric connection through a mercury well operates when the beam drops, giving a bell signal for the start and end of each run. This device has proved very convenient and accurate. One-half pound of gasoline is regularly used for light loads and one pound for larger loads, giving runs of from two to six minutes' duration.

A hand air-pump is attached to the weighing tank, giving the necessary pressure to supply fuel to the carbureter.

Radiator and Exhaust

Since the car is at rest and only the motor, transmission, and rear wheels are in motion, the radiator is deprived of the active air circulation found on the road. To prevent overheating the cylinders a supply of cooling water is added to the radiator, with the overflow of hot water running to waste. The temperature of the escaping water is recorded and is usually kept at 160° F.

Proposed Modifications

Steps are now being taken to increase the convenience of the operator of the measuring apparatus, without changing the

methods used. It is proposed to have both the revolution-counter for the traction drums and the time-clock connected electrically with the scale-beam for gasoline weighing. In this way both these records will be determined without personal error of the observers. A recording dynamometer is also planned to give a record of the load on the scales, to show the constancy of the draw-bar pull. This will be used to supplement, not to replace, the accurate weighing system in use.

A powerful fan, driven at variable speeds, blowing air at the radiator, is also planned. This may obviate the need of water overflow for cooling the radiator and may also make possible the observation of full loads at higher speeds.

Method of Testing

Rolling Resistance

In calculating the rolling resistance the following steps are taken:

(a) The tires are pumped to 70 pounds pressure.

(b) The car is towed on level floor, in high gear, declutched, to obtain the pull by dynamometer.

(c) The projected area of the car-body is measured, the width across the mud-guards and the height from the running-board to the top of the wind-shield (half up), or to the top of closed cars. Allowance is made for stream-line bodies by reducing the area slightly.

If the slow pull is denoted by r, the projected area by a, and the speed in miles per hour by S, the rolling resistance is calculated by the formula

$$R = r + .003\ a\ S^2 \tag{28}$$

This formula is assumed to give the rolling resistance, in pounds, of the car at various speeds on smooth, level road, comparable to the laboratory floor.

Description of Runs

After the car is placed on the drums and the motor warmed up by a preliminary trial, the following runs are made, during

which careful measurements are taken of the load on the scales, speed, gasoline weight, temperature of cooling water, and time of run:

Run 1. At 5 miles per hour, or slowest speed possible, load equal to rolling resistance, level.

Run 2. At 10 miles per hour, load equal to rolling resistance, level.

Run 3. At 10 miles per hour, load maximum at that speed.

Run 4. At 20 miles per hour, level road resistance.

Run 5. At 20 miles per hour, maximum load.

Run 6. At 30 miles per hour, level road resistance.

Run 7. At 30 miles per hour, maximum load.

Run 8. At 40 miles per hour, level road resistance.

Run 9. At 40 miles per hour, maximum load.

Measurements are made during the nine runs and recorded on suitable log sheets.

Diagram of Results

From these records calculations are made in two groups, one for level road conditions and one for maximum load, both over the entire range of speed covered. These calculations are made for

Actual speed, miles per hour S.

Maximum draw-bar pull D.

Level road draw-bar pull R.

Gasoline, reduced to miles per gallon, level road f.

Gasoline, reduced to miles per gallon, full road F.

Effective draw-bar pull $(D - R) = P'$.

Horse-power at rear tires, level road (calculated from R) Y.

Horse-power at rear tires, full load (calculated from D) Z.

These results are plotted and curves drawn as shown in Chart V.

The curves drawn through the plotted points are in three groups and are subject to a check. The draw-bar curves for level road and full load conditions will intersect at the maximum

speed of the car. The horse-power curves, *Y*, *Z*, will intersect at the same speed as given by the first-mentioned curves.

The fuel curves for miles per gallon of gasoline are plotted

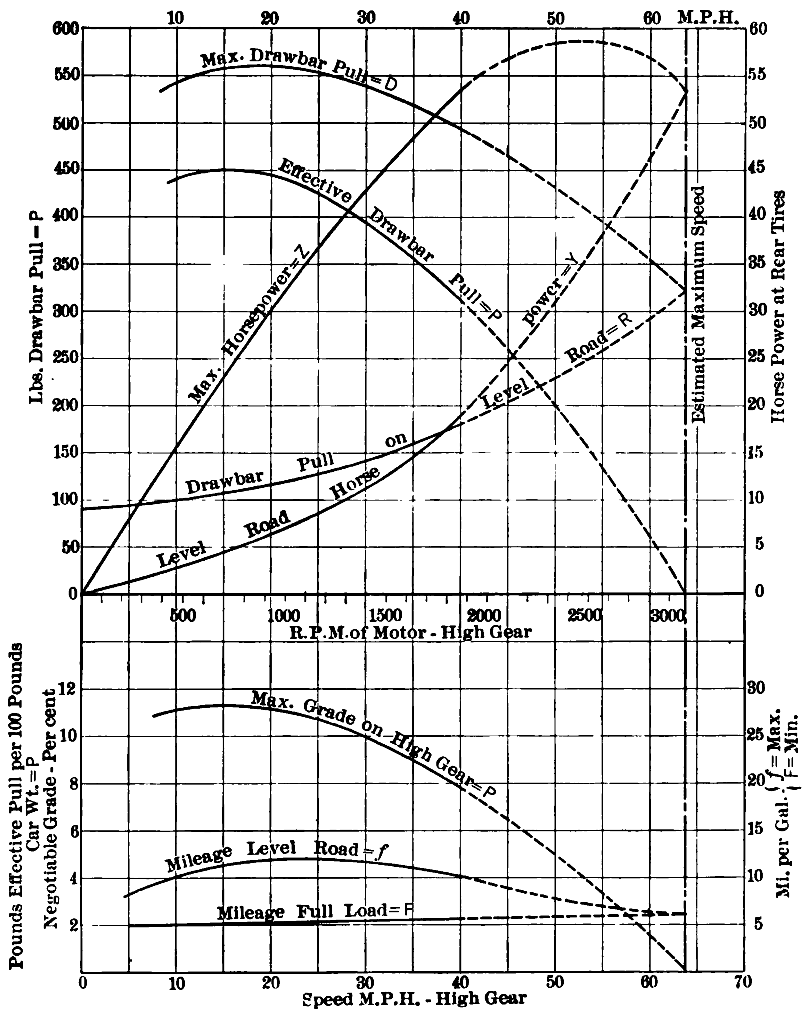


CHART V.

below, for clearness, using the same abscissæ. These curves likewise will intersect at the point of maximum speed.

After smoothing out the curves through the plotted points, the exact values of the draw-bar pull, horse-power, and miles per gallon can be read off at any intermediate speeds with greater accuracy than the original plotted points.

TABULAR REPORT FORM

DATA FROM TEST OF

......................................Model..................

For..

Made at..............................Date...................

By...

DIMENSIONS OF CAR

Wind-resisting area (a).........sq. ft. Wt. with driver...............lbs.

Rolling resistance declutched.....lbs. Drive...........................

Gear ratio, 1st........2d........... 3d..........4th................

Tires, size, front .... Make...........................

Tires, size, rear .... Make...........................

Tires, tread, front .... Inflation.....................lbs.

Tires, tread, rear .... Inflation.....................lbs.

Ignition.......................... Carbureter......................

Fuel sp. gr............at.........F. Wt. per gal..................lbs.

POWER AND FUEL MEASUREMENTS

	Draw-bar Pull			Fuel Data			Horse-power at Rear Tires	
S	R	D	P	f	F	A	Y	Z
Speed of Car $\frac{60N}{301T}$ Miles per Hour	On Level Road $r + .003\ aS^2$ Pounds	Max. at Full Load $1.234L + 35$ Pounds	Net Effective Pull $\frac{(D-R)\ 100}{W}$ Pounds	On Level Road $\frac{w\ ST}{60p}$ Miles per Gallon	At Full Load $\frac{w\ ST}{60p}$ Miles per Gal.	Acceleration $.322P$ Feet per Sec. per Sec.	Level Road $\frac{RS}{375}$ H. P.	Full Load $\frac{DS}{375}$ H. P.
5								
10								
20								
30								
40								
50								

Explanation

N = revolutions of drums during run from records.
T = time of run, minutes, from records.
r = rolling resistance, slow speed, by dynamometer, lbs.
L = load on brake-scale beam during run, lbs.
W = weight of car, including driver, lbs.
w = weight of one gallon of gasoline, lbs.
p = lbs. of gasoline used during run from records.
a = wind-resisting area of car-body, measured.
S = speed of car in miles per hour.

Expression of Results

As noted in the introduction to this paper, a test of performance should be of value to the commercial as well as to the technical end of the automobile industry. In order to be intelligible to other than the trained engineer, results should be expressed in non-technical terms of common usage. At the same time the expression of results must omit no detail of desirable information.

On close inspection these conditions become less difficult than they at first appear. The satisfaction of a motor-car owner is dependent upon four factors, so far as performance is concerned:

First, the car must have a wide speed-range on the high gear; second, it must accelerate quickly; third, it must possess suffi-

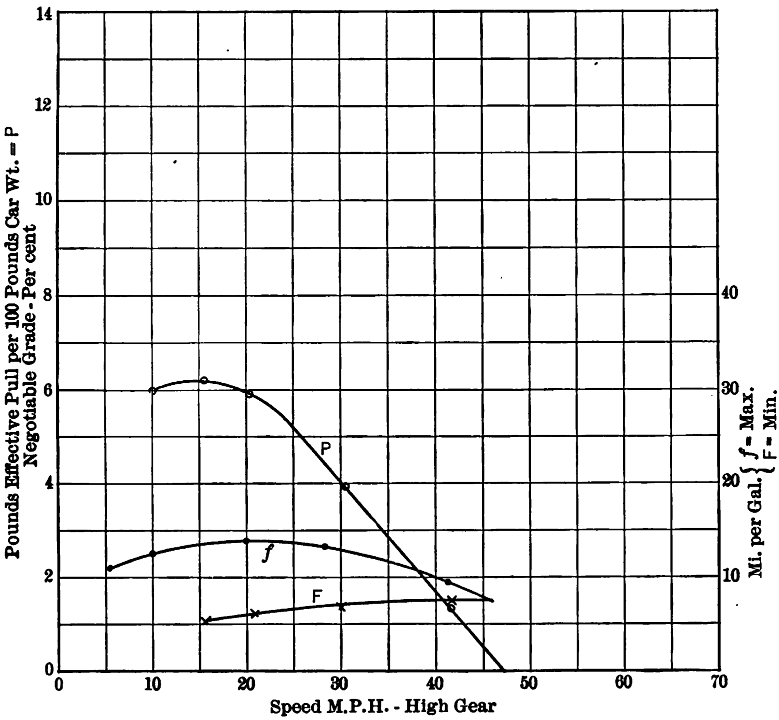


CHART VI.

Car No. 1. 1913 Roadster. 6 cylinders. Bore, 4 inches. Stroke, 5½ inches. Weight, with driver, 4,435 pounds. Rolling resistance, 80 pounds. Wind-resisting area, 21.8 square feet. Tires, 37 x 5, non-skid. Inflation, 70 pounds rear, 60 pounds front.

cient power to negotiate grades, or to overcome heavy road conditions; fourth, it must be economical of fuel.

Acceleration and Hill-Climbing Ability

The second and third factors are direct functions of the excess power of the car. By "excess power" is meant the total effort of the engine minus the total rolling resistance. In other words, it is the excess of pull of which the car is capable at any

speed, exerted on the roadway, over and above the pull necessary to move the car against its own rolling resistance at that speed.

In brief, it is the net effective power of the car and may be conveniently expressed in pounds pull and designated by P'. Concretely, P' is determined, as already noted, by subtracting the total rolling resistance from the maximum draw-bar pull, as determined by the methods herein outlined.

If the net effective power P' be considered as force, we have

$$P' = MA$$

where

$$M = \text{Mass or } \frac{W}{g}$$

then if

A = acceleration in feet per sec. per sec.
W = weight of car (with driver) in pounds.
G = per cent grade that can be surmounted.

$$A = \frac{32.2P}{W}$$

$$G = \frac{100\,P}{W} \qquad (29)$$

If, therefore, we reduce P' to P, which equals the net effective power per 100 pounds of car weight, the same scale gives a direct reading of the maximum gradient a car will surmount at a given speed, while

$$A = 0.322\,P \qquad (30)$$

Speed Range

Plotting the curve of net effective power per 100 pounds of car weight with pounds for a unit, as ordinates and speed, in miles per hour, as abscissæ, the point where the P curve falls on the zero line of power establishes the maximum speed of the car. The minimum speed at full load is designated by the opposite end of the P curve, while the minimum speed on level road is shown by the left-hand of the f curve, both as established by observation during the test.

Fuel Consumption

From the foregoing it is seen that *a single curve expresses three of the four desired factors.* There remains only fuel con-

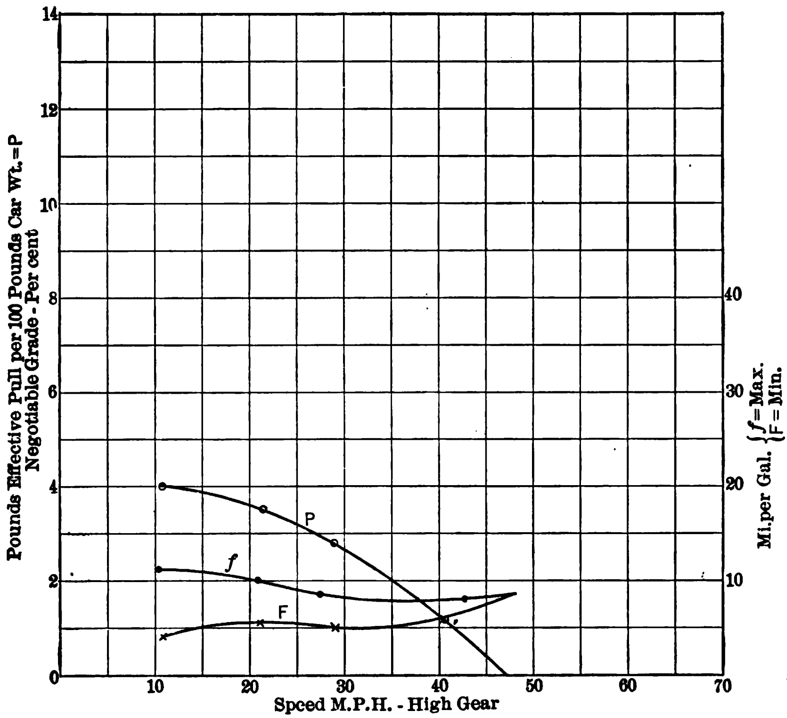


CHART VII.

Car No. 2. 1915 Touring Car. 6 cylinders. Bore, 4 inches. Stroke, 5½ inches. Weight, with driver, 4,950 pounds. Rolling resistance, 83 pounds. Wind-resisting area, 22.6 square feet. Tires, 37 x 5, plain tread. Inflation, 70 pounds front and rear. Gear ratio, direct drive, 3.53.

sumption. As this item covers a range from the lightest to the heaviest loads, it seems best to plot both extremes.

This is conveniently done on the same chart by renumbering the ordinates on the right of the diagram, using as a standard the common unit of miles per gallon. Minimum fuel consumption then becomes maximum mileage and is, of course, the mileage possible on level cement road. This may be designated by *f*. The mileage at full load may be designated by *F*.

Fuel Check on Speed Limit

As already noted, it is evident that maximum speed on level road is identical with full load at the same speed. It is therefore

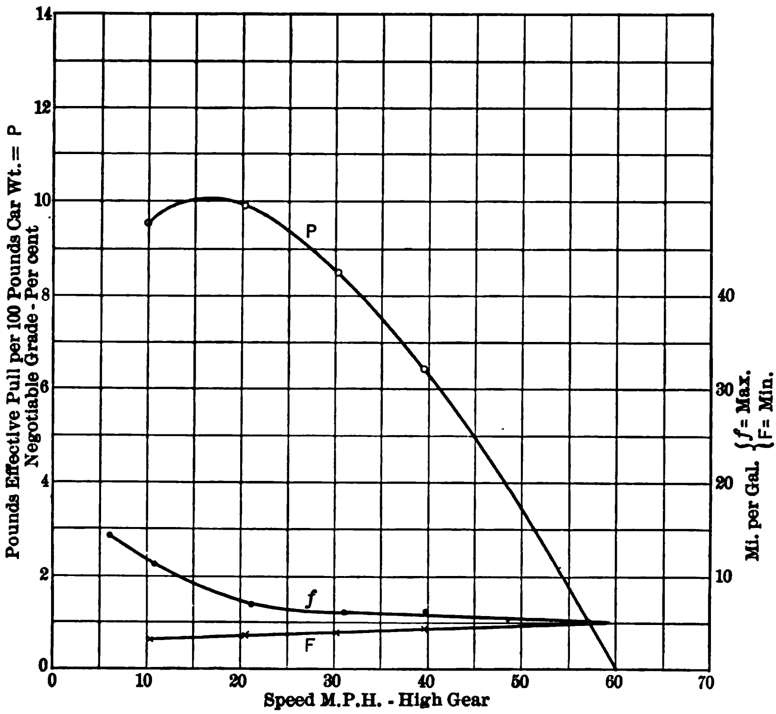


CHART VIII.

Car No. 3. 1915 Touring Car. 6 cylinders. Bore, 4 inches. Stroke, 5½ inches. Weight, with driver, 4,562 pounds. Rolling resistance, 40 pounds. Wind-resisting area, 19.2 square feet. Tires, 36 x 4½, cord. Inflation, 70 pounds front and rear. Gear ratio, direct drive, 3.78.

clear that the f and F curves should join on the same abscissæ where the P curve reaches a zero value. This affords a positive check on the accuracy of the observations and plotting. Charts VI to XI show the efficacy of this check and its corrective influence on the characteristics of all curves.

The influence of the termination of the curves is clearly shown in Chart VIII. The last observation on the *f* curve indicates a maximum speed somewhat higher than is shown in the diagram,

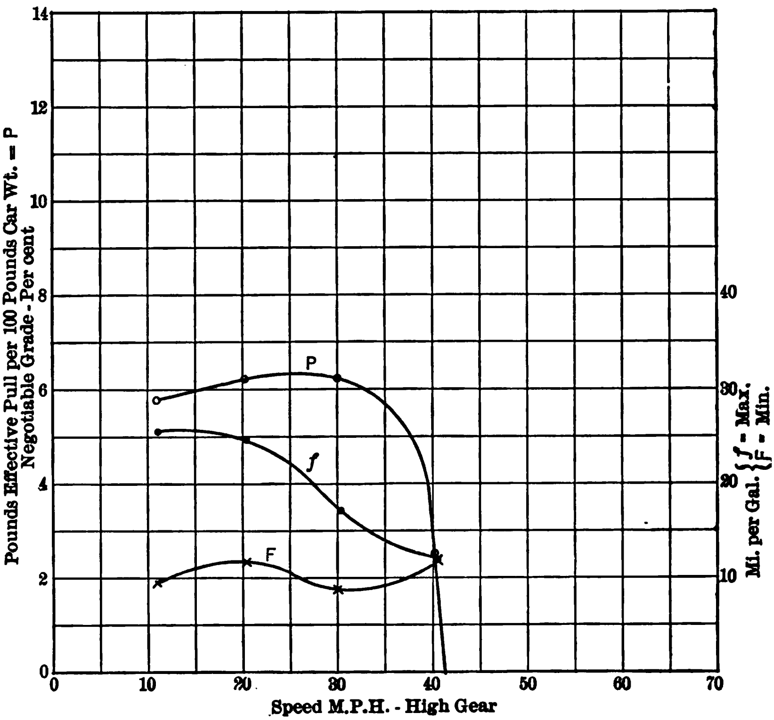


CHART IX.

Car No. 4. 1915 Touring Car. 6 cylinders. Bore, 3⅝ inches. Stroke, 4 inches. Weight, with driver, 3,020 pounds. Rolling resistance, 45 pounds. Wind-resisting area, 20.2 square feet. Tires, 34 x 4½, cord. Inflation, 70 pounds front and rear. Gear ratio, direct drive, 3.71.

whereas the *P* curve plots smoothly to the speed limit shown. This discrepancy was doubtless due to manual adjustment of the carbureter by the dash control.

Chart IX shows two actual observations of fuel, falling practically together on a point in close agreement with the speed limit as determined by the *P* curve.

Speed Limit of Observations

In the diagrams shown herewith, but one, Chart X, is incomplete through lack of additional observations at higher speeds. On account of instances like this it is desirable to determine points on the curves at as high vehicle speeds as possible. Usually it is inexpedient to run the car, particularly at full load, at speeds exceeding 40 miles per hour because as the car is stationary there is a tendency to overheating through the absence of the cooling effect of the motion of the car on the road. Ordinarily, however, a sufficient number of observations may be taken at and below 40 miles per hour to establish reasonable accurate projection of the curves.

Applicability to Road Conditions

The true value of this method of testing depends largely upon the fidelity with which its results can be duplicated on the road. To establish this, several road checks have been conducted by the authors under strictly test conditions, and by others under ordinary conditions of driving. For example, a four-cylinder car was driven over a practically level course of 2,801 feet on Orange Street, New Haven, a road surface corresponding closely to that of the laboratory floor. The throttle was set in various marked positions and the speed accurately noted by a stop-watch. Tests were duplicated with the car driven in both directions to eliminate the effect of any possible slight grade. The car was then placed on the test stand and the throttle opened to the same positions. Following is a tabulation of the results:

Comparison of Test-stand and Road-test Results

Speed		Draw-bar Pull		
On Road M. P. H.	On Test-Stand, M. P. H.	By Formula, Pounds	Actual, Pounds	Error, Per Cent
14.83	14.27	57.6	58.0	0.7
18.84	18.23	65.2	66.5	2.0
23.00	22.23	75.1	76.7	2.12
26.90	27.75	86.2	88.0	2.09
30.40	30.95	97.6	102.8	5.33

Car No. 1 of the present series of tests was driven 90 miles by the owner on selected macadam roads and fell but 0.6 of a mile per gallon below the average shown by the block test between the same speed limits.

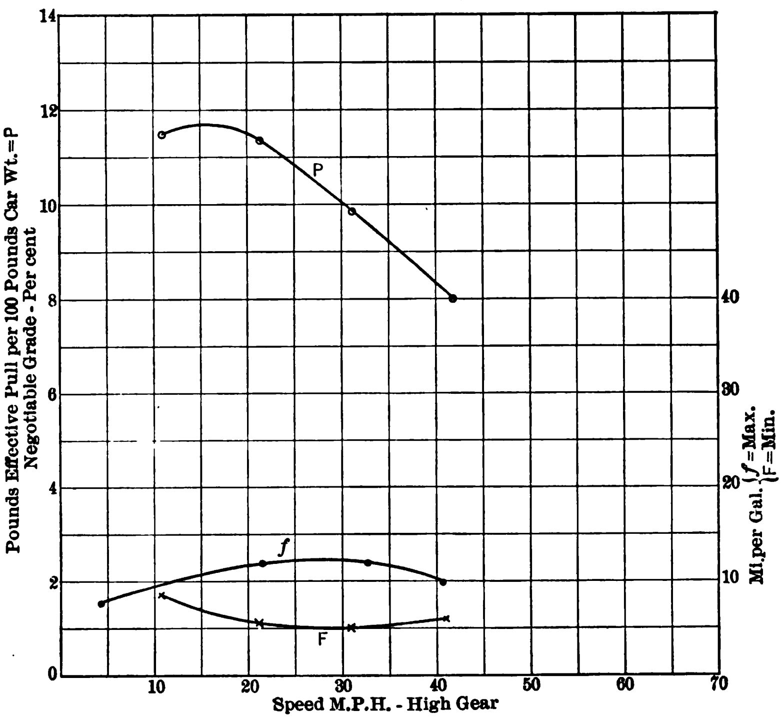


CHART X.

Car No. 5. 1915 Touring-Car. 8-cylinder. Bore, 3⅛ inches. Stroke, 5⅛ inches. Weight, with driver, 4,020 pounds. Rolling resistance, 78.5 pounds. Wind-resisting area, 18.6 square feet. Tires, 36 x 4½, non-skid. Inflation, 70 pounds front and rear. Gear ratio, direct drive, 5.02.

Another car, showing 11.2 miles per gallon under test, was actually driven 11.1 miles over a selected road with a carefully weighed gallon of gasoline. This duplication of test-stand results by different drivers on different roads must be considered as something more than coincidence, and its testimony lends weight to the accuracy of the method.

Disclosure of Characteristics

This method also shows to a surprising degree the relative action of certain parts of different cars. If the car is equipped with manual spark control and dash control of the carbureter, it also shows the relative skill of different drivers and the effect

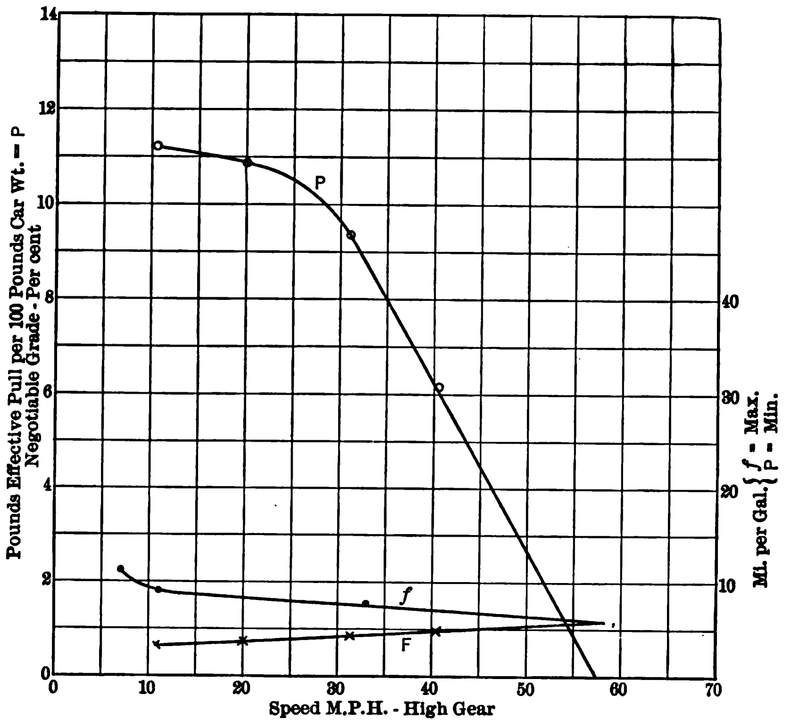


CHART XI.

Car No. 6. 1915 Touring-Car. 6-cylinder. Bore, 4½ inches. Stroke, 5½ inches. Weight, with driver, 5,020 pounds. Rolling resistance, 47 pounds. Wind-resisting area, 22.3 square feet. Tires, 37 x 5, cord. Inflation, 70 pounds, front and rear. Gear ratio, direct drive, 3.5.

of these adjustments in the hands of the average user may be learned therefrom.

The test shows the performance of the car *as it is* at the moment of testing. What difference another make of tires, a different adjustment of the carbureter, or the change of any

other feature would make can be determined only by repetition of the test under the new conditions. For instance, in Chart IX, the driver evidently desired his car to establish a reputation for

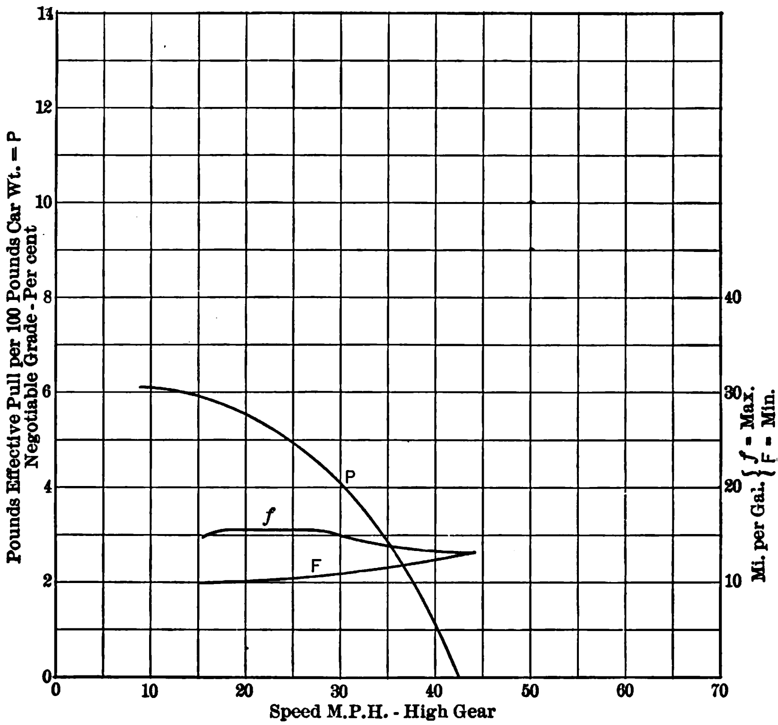


CHART XII.

Renault Touring-Car. Plotted from test by Dr. A. Riedler (see Chart XIII).

fuel mileage. He was successful, but at the expense of speed, acceleration, and hill-climbing ability.

Again, certain characteristics of the carbureter are clearly shown. For instance, the performance of the carbureter in Car No. 1 (Chart VI) was wholly consistent, giving smooth curves at all speeds, a fuel consumption at full load directly proportional to the speed, and a maximum mileage on level road at between 20 and 25 miles per hour.

Car No. 3 (Chart VIII) shows equally good action at full

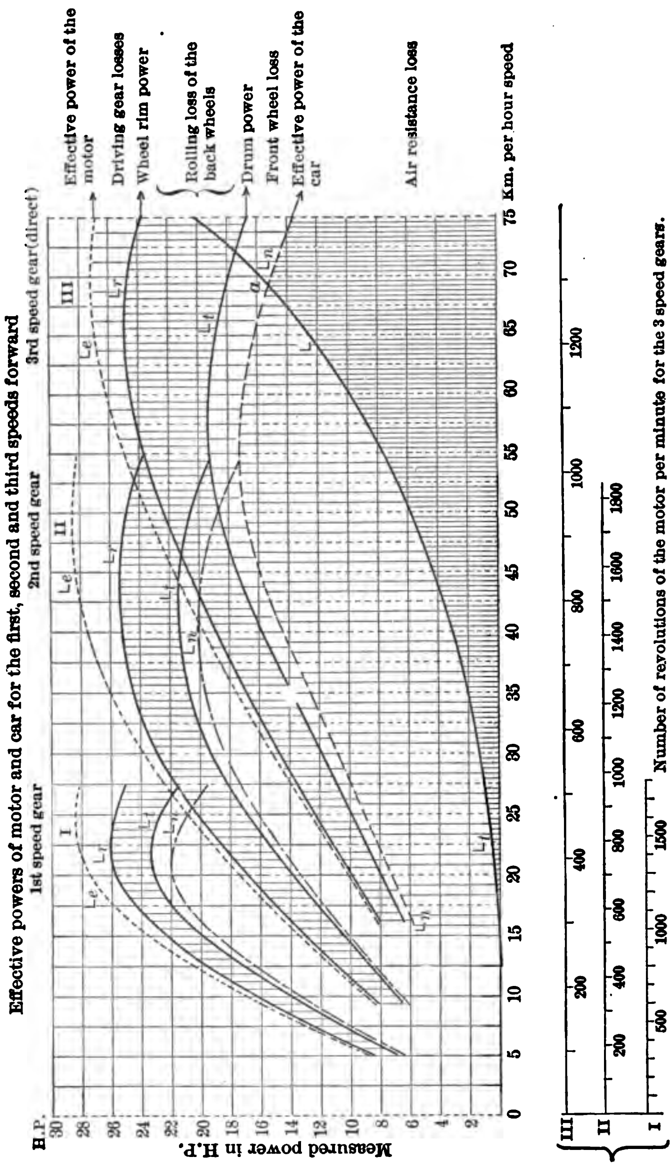

Effective powers of motor and car for the first, second and third speeds forward
1st speed gear
2nd speed gear
3rd speed gear (direct)
I
II
III
Effective power of the motor
Driving gear losses
Wheel rim power
Rolling loss of the back wheels
Drum power
Front wheel loss
Effective power of the car
Air resistance loss
Km. per hour speed
H.P.
Measured power in H.P.
Number of revolutions of the motor per minute for the 3 speed gears.

CHART XIII.

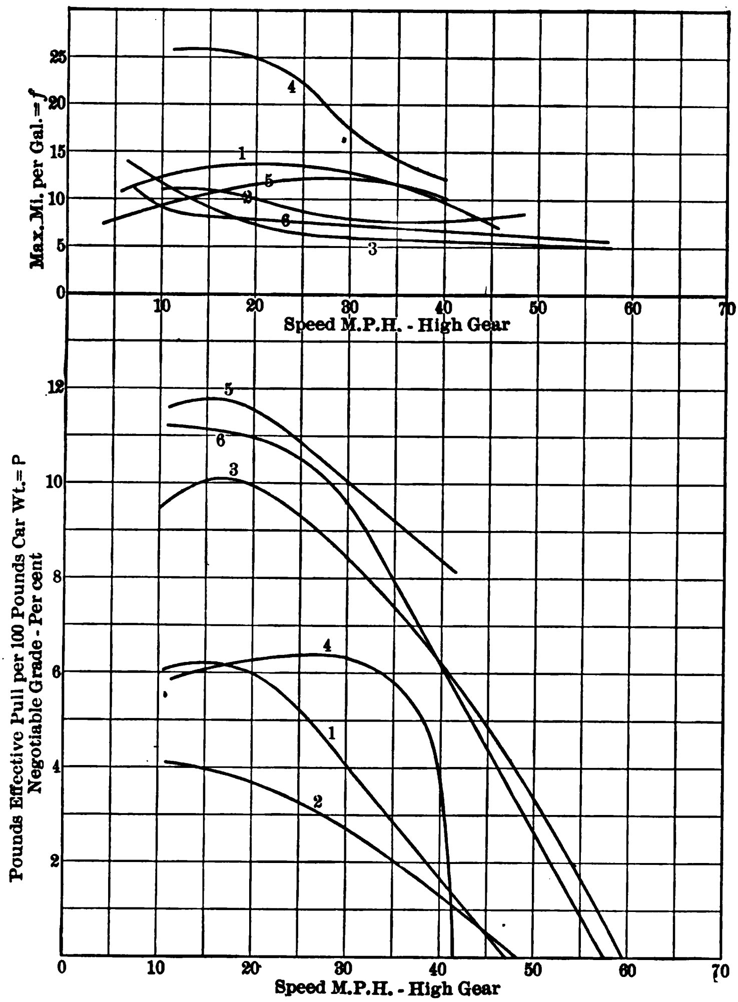


CHART XIV.

Comparison of fuel consumption and power of six representative American cars. Cars Nos. 1 to 6, inclusive.

load, but faulty compensation under throttle, with resulting maximum mileage at minimum speed. That this is really a characteristic of the carbureter is shown by comparing the same make of carbureter on a different car, Car No. 6 (Chart XI), which exhibits the same pronounced characteristics.

Note also the similarity of the character of both fuel curves in Car No. 2 (Chart VII), and compare them with similar characteristics developed by a different carbureter on Car No. 4 (Chart IX).

Again, the rolling resistance of cars of approximately the same weight is found to vary markedly. Whether this is due to different tires or to internal friction can only be determined by substituting in one case, or by more detailed investigation in the other.

The Performance Test as a Basis for Subsequent Investigation

The possibilities of this method for maintaining constant all conditions, except the one under investigation, offer alluring opportunity for the investigation of various components entering motor-car construction. The development of the method of testing herein outlined has, of itself, occupied so much of their time, that the enticing field of detailed analysis of the results has hardly been entered by the authors.

Suggestions concerning more detailed investigation are outside the province of this paper, but, in illustration of the possibilities, Chart XII is an expression, by the proposed method, of a test of a Renault touring-car made by Dr. A. Riedler, of Berlin, Germany. Chart XIII is a reproduction of Dr. Riedler's complete test as published in a translation of his work entitled "The Scientific Determination of the Merits of Automobiles." In this diagram Dr. Riedler has shown the sources of loss from different causes, expressed, as the authors believe, unfortunately in terms of horse-power. Chart XII may be said to indicate an effect, while Chart XIII is an analysis of the cause. The facts of Chart XII may be accurately determined in two hours,

while those of Chart XIII can be learned only after the most protracted and painstaking effort.

Draw-Bar Pull vs. Horse-Power

The authors can not close without recording a protest against the use of horse-power as a unit for motor-car rating. Whatever may be its value in the classification of motor-car engines, it seems utterly inconsistent to apply it to the performance of a vehicle. *It is the pull or push* of the tire on the road that is effective in the propulsion of a car. Witness the utter absurdity of a steam-car equipped with a 20 horse-power engine, outpacing and outclimbing gas-cars, the engines of which will develop upward of 80 horse-power on the block. The steam-car accomplishes this by greater and more uniform torque (or turning-moment) delivered to its rear wheels through the continued and overlapping admission of high cylinder pressures; therefore, it is clearly this torque, or turning effort, that should be recognized, and its direct and easily measurable result, draw-bar pull, seems to be the logical, final unit of such measurement.

CHAPTER V

DIRECT DETERMINATION OF CARBURETER ACTION

As HAS been shown in preceding chapters, the primary function of a carbureter is to maintain the relative proportions of gas and air in an explosive mixture. The direct determination of how well this function is performed is attended with difficulties.

The amount of fuel entering the mixture may be accurately measured by ordinary means. The air content is by no means so easy of determination. The problem is complicated by the fact that in the internal combustion engine the air-flow is induced by a series of more or less separate impulses, so that the pressure flow is pulsating in character. The result of this is to introduce inertia effects and other influences, which react on the velocity of the flow to such an extent as to make its accurate determination exceedingly difficult.

THE ANEMOMETER

The anemometer, or other form of mechanical meter, is not sufficiently responsive to the frequent pulsations even if the errors inherent in such instruments could be tolerated.

ORIFICE IN THIN PLATE

Attempts have been made to measure the flow through known orifices in thin plate into a chamber in which pressures are indicated by means of a manometer. The chief difficulty experienced with this apparatus is the determination and maintenance of the actual coëfficient of flow. This varies with the size of the orifice and with the pressure, density, and velocity of the air—all variable conditions.

Formula of Flow

Durley, Trans. A.S.M.E., Vol. XXVII, page 193, shows that if

w = weight of gas discharged per second in lbs.
P_1 = pressure inside orifice in lbs./sq. foot.
P_2 = pressure outside orifice in lbs./sq. foot.
γ = ratio of specific heat at constant volume to that at constant pressure.
d = diameter of orifice in inches.

then for air at 60° F. when $\gamma = 1.404$

$$w = 0.000491\, d^2 P_1 \sqrt{\left(\frac{P_2}{P_1}\right)^{1.425} - \left(\frac{P_2}{P_1}\right)^{1.712}} \qquad (31)$$

If we neglect the changes of density and temperature occurring as the air passes through the orifice we obtain a simpler, though approximate formula for the ideal discharge.

$$w = 0.01369\, d^2 \sqrt{\frac{iP}{T}} \qquad (32)$$

in which

d = diameter of orifice in inches.
i = difference in pressure measured in inches of water.
P = mean absolute pressure in lbs./sq. ft.
T = absolute temperature in F.° = F.° + 461.

Thickness of Plate

Up to pressures of about 20 inches of water, the results of the foregoing formulæ agree very closely. At higher differences of pressure divergence becomes noticeable. The values found by these formulæ are to be multiplied by a coëfficient c, determined experimentally. They hold good only for orifices of the particular form experimented with and bored in plates of the same thickness, viz.: iron plates 0.057 inches thick.

Necessary Conditions

Experiments and curves plotted from them indicate that up to a pressure of about 20 inches of water

(1a) The coëfficient for small orifices increases as the head increases.
(1b) For a 2-inch orifice, the coëfficient is almost constant.
(1c) For orifices larger than 2 inches, the coëfficient decreases as the head increases and at a greater rate the larger the orifice.
(2) The coëfficient decreases as the diameter of the orifice increases and at a greater rate the higher the head.
(3) The coëfficient does not change appreciably with temperature between 40° and 100° F.
(4) The coëfficient (at heads under 6 inches) is not appreciably affected by the size of the box in which the orifice is placed, if the ratio of the areas of the box and orifice is at least 20: 1.

TABLE I

Coëfficient of Discharge (c) for Various Heads in Inches of Water and Diameters of Orifice in Inches, in Plate 0.057 Ins. Thick.

Diameter of Orifice	1-Inch Head	2-Inch Head	3-Inch Head	4-Inch Head	5-Inch Head
5/16	0.603	0.606	0.610	0.613	0.616
½	0.602	0.605	0.608	0.610	0.613
1	0.601	0.603	0.605	0.606	0.607
1½	0.601	0.601	0.602	0.603	0.603
2	0.600	0.600	0.600	0.600	0.600
2½	0.599	0.599	0.599	0.598	0.598
3	0.599	0.598	0.597	0.596	0.596
3½	0.599	0.597	0.596	0.595	0.594
4	0.598	0.597	0.595	0.594	0.593
4½	0.598	0.596	0.594	0.593	0.592

Apparatus for Carbureter Measurements

For purposes of carbureter measurements orifices of various diameters may be bored in a plate 0.057 inches thick, forming one side of a closed box. Provision should be made to close all orifices but the one in use. In accordance with provision 4 of the preceding paragraph, the cross-sectional area of this box should be at least twenty times the area of the largest orifice.

Rubber Diaphragm

One side of the box is made of sheet rubber, the flexibility of which aids materially in neutralizing the pulsations of the air-current. The box is provided with a thermometer and is connected to one leg of a manometer graduated in inches and tenths. Connection is made from the box to the carbureter by means of suitable piping. If the carbureter has more than one intake opening it is well to inclose the entire instrument in an air-tight box and connect this box to the meter-box by a pipe.

The carbureter-box may be made of sheet metal with one side acting as a cover, secured in place against an air-tight gasket. The box is supported between the carbureter and manifold flanges, being bored to register with manifold passage and with the cap screws securing the flanges. Tightness is secured by gaskets between both flanges and the box.

It seems unnecessary to add that every precaution must be taken to guard against air leakage with the apparatus, and to this end all joints must be made air-tight.

With an apparatus so constructed, the weight of air used in the carbureter may be determined by a derivation from Durley's formulæ, substituting observed values in the following equation:

$$w = 0.01369\, c\, d^2 \sqrt{\frac{m\,(70.748\, B - 5.184\, m)}{T}} \tag{33}$$

when

c = a coëfficient selected from Table I.
w = weight of air used in lbs./sec.
d = diameter of orifice in inches.
m = manometer reading in inches of water.
B = barometric pressure of the atmosphere in inches of mercury.
T = absolute temperature of the air = F.° + 461.

Orifice Diameters

The diameter of the orifice should be selected with a view to maintaining a manometer reading sufficiently high at the lowest

engine speeds to insure accuracy of observation. As the speed increases, larger diameters should be employed so the head of water is kept as low as is consistent with convenient manipulation.

Objection

One of the principal objections to this method is that the carbureter is at all times operating at sub-atmospheric pressures, a condition which may not be fairly comparable to its operation in actual service. This error may be materially reduced, however, by employing orifices which give low readings on the manometer, substituting larger orifices as the demand for air increases.

This manipulation furthermore reduces the error in the coëfficient of flow, which may be easily kept within 1 per cent.

The Venturi Meter

Principles Involved

Probably the most practical method for the direct measurement of air is by the use of the Venturi meter. This instrument depends for its action on the loss of head caused by the increased velocity of flow through a constriction in the cross-sectional area of a tube. By equation (4) (Chapter I), this loss of head, h, is

$$h = \frac{Va^2}{2g}$$

therefore, by a measurement of the loss of head, h, we are enabled to determine the velocity by the formula

$$Va = c\sqrt{2gh}$$

when c = a coëfficient of flow.

When the construction of the tube is made with highly finished surfaces at angles which closely follow the natural contraction of the vein of flow, this coëfficient is nearly constant at greater than 0.98.

The head or pressure difference is measured in inches of water

by connecting one leg of a manometer U tube with the "upstream" end of the meter, while the other is connected to the throat.

Calibration

The manufacturers of these meters furnish a calibration curve of each instrument, showing the actual discharge in cubic feet per minute under standard conditions of temperature and barometer, viz., 62° F. and 29.92 inches of mercury.

Barometric and Temperature Correction

For any other temperature and pressure, the discharge may be determined by the formula

$$V = 0.24 \quad M \frac{T}{B} \sqrt{\frac{B}{T}} \tag{34}$$

when

V = volume of air in cubic feet per minute.
M = meter reading in cubic feet per minute.
T = absolute temperature of the atmosphere = (F.° + 459).
B = barometer reading in inches of mercury.

As air/gas ratios are usually given by weight, the weight of air in pounds per minute (W) is found by the following:

$$W = 0.31835 \, M \sqrt{\frac{B}{T}} \tag{35}$$

Application to Carbureter Measurements

As is the case with the orifice in thin plate, it is desirable to neutralize pulsations in the air-current by a flexible diaphragm forming one side of a box into which the air is metered and from which it is withdrawn to the carbureter.

The use of at least two sizes of meters is also desirable: one with a throat diameter about 0.5 inches, and the other with a throat diameter of from 2 to 3 inches. The former indicates from 4 to 26 cubic feet per minute, with from 0.5 to 18 inches difference in the head of water in the U tube, while the latter discharges

25 to 200 cubic feet per minute, with manometer deflections of from 0.2 to 16 inches. By substituting one instrument for the other, accurate measurement may be secured over a range sufficient to cover small engines at slowest speeds or large engines at highest speeds.

CHAPTER VI

CHEMISTRY OF CARBURETION

Complexities

The performance of an automobile engine presents problems of a physicochemical nature. Because of the complexities of the interrelationship of these two branches of science, the chemical investigation of combustion reactions and their physical effect on power output has registered less progress than its importance deserves. This seems due in large measure to a lack of understanding coöperation between chemist and physicist, and it is therefore gratifying to note the increasing interest shown by engineers in the study of exhaust gases.

Primarily, power is developed in the internal combustion engine as the direct effect of heat liberated solely by means of certain chemical reactions known as combustion. Knowledge of these reactions is, therefore, of prime importance. Because combustion takes place, after a fashion, throughout such a wide range of mixture composition, engineers are prone to lose sight of the necessity for a careful study of these fundamental reactions, even though they form the very basis of power development.

Availability of Exhaust Gas Analysis

Analysis of the exhaust gases from an internal combustion engine furnishes one of the most convenient methods of comparing carbureter performances. It must be admitted that gas analysis has not yet reached the point where complete information may be obtained, but with the well-known methods in common use sufficient data may be obtained which, when properly interpreted, will be found wholly consistent and in point of fact sufficiently accurate for all practical purposes.

Before the true value of gas analysis can be fully realized,

an intimate knowledge of the chemical and thermal reactions taking place within the cylinder is necessary.

COMBUSTION

The fuel used in automobile engines is a hydrocarbon, or really a combination of several hydrocarbons forming part of what is known as the paraffin series. These liquid distillates, obtained from crude petroleum, have the general chemical formula C_nH_{2n+2}, which may be explained as a substance composed of n molecules of carbon and $2n + 2$ molecules of hydrogen. Gasoline, for example, is composed largely of hexane, which contains 6 molecules of carbon, namely, $n = 6$ combined with $2 \times 6 + 2 = 14$ molecules of hydrogen $= C_6H_{14}$.

Reactions of Hexane

The term combustion may be defined as the union of a substance with oxygen. Both hydrogen and carbon, when raised to the required temperature, in the presence of air unite very readily with the oxygen in the air, the hydrogen forming water and the carbon, carbon-dioxide, CO_2, when the proper quantity of air is present, or carbon monoxide, CO, when there is insufficient air for complete combustion. These reactions may be expressed as follows, if we assume gasoline to be composed entirely of hexane:

$$2C_6H_{14} + 19O_2 + 71.3N_2 = 12CO_2 + 14H_2O + 71.3N_2 \quad (36)$$

This equation shows perfect combustion in which all the carbon is oxidized to CO_2 and all the hydrogen has formed water.

The following equation presupposes an insufficient amount of air and its consequent imperfect combustion:

$$2C_6H_{14} + 17.5O_2 + 65.8N_2 = 9CO_2 + 3CO + 14H_2O + 65.8N_2 \quad (37)$$

Here the oxidation of the carbon has been incomplete, resulting in the formation of both CO_2 and CO.

Relative Volume of the Exhaust

From equation (36) we find that 2 volumes of hexane unite with 90.3 volumes of air and that the exhaust gas occupies

$12 + 14 + 71.3 = 97.3$ volumes. As the 14 volumes of H_2O, existing in the exhaust as steam, promptly condense, the final exhaust consists of 83.3 volumes, composed of

$$CO_2 = \quad 14.4 \text{ per cent.}$$

$$N_2 = \frac{85.6}{100.0} \text{ " \quad " }$$

It must be understood that equations (36) and (37) do not represent exactly what happens. Gasoline is of highly complex composition, rarely containing more than 85 or 90 per cent of hexane, the remainder being compounds of uncertain and complex composition. This renders exact determinations of the combustion reactions almost impossible. It is highly probable, also, that the process of combustion is by no means so direct as the foregoing equations would indicate. They are given here merely as the basis for an understanding of the process by which products of combustion are formed. They serve also to illustrate the character and composition of the exhaust gases from complete and incomplete combustion.

Chemical Composition of Air

At 32° F. air contains

	By Weight	By Volume
Oxygen	23.6 per cent.	21.3 per cent.
Nitrogen	76.4 " "	78.7 " "
	100.0 " "	100.0 " "

Therefore, a given quantity of air weighs

$$\frac{100}{23.6} = 4.23 \text{ times its oxygen content} \tag{39}$$

or

$$\frac{100}{76.4} = 1.31 \text{ times its nitrogen content.} \tag{40}$$

Similarly, a given quantity of air will occupy

$$\frac{100}{21.3} = 4.69 \text{ times the volume of its oxygen} \tag{41}$$

or $$\frac{100}{78.7} = 1.27 \text{ times the volume of its nitrogen.} \qquad (42)$$

Determination of Air Necessary for Combustion

The amount of air necessary for combustion may be determined as follows:

Let it be required to determine the amount of air necessary for the combustion of one pound of carbon.

First write the combustion equation

$$C + O_2 = CO_2 \qquad (43)$$

Substitute atomic weights

$$12 + 32 = 44 \qquad (44)$$

Elements

Divide O by C

$$\frac{32}{12} = 2.66 \text{ lbs. of O} \qquad (45)$$

That is, 1 pound of carbon requires 2.66 pounds of oxygen for its complete combustion. As, by the preceding paragraphs, air weighs 4.23 times its oxygen content, 2.66 pounds of oxygen will be equivalent to

$$2.66 \times 4.23 = 11.28 \text{ lbs. of air.}$$

At 32° F. one pound of air occupies 12.387 cubic feet, so that the volume of 11.28 pounds of air will be

$$11.28 \times 12.387 = 139.2 \text{ cu. ft. of air.}$$

At any other temperature the volume will be proportional to the absolute temperature

$$v = \frac{V \times T}{t} \qquad (46)$$

Temperature Correction

Thus at 90° F. the volume required would be

$$\frac{139.2 \times (459 + 90)}{(459 + 32)} = 155 \text{ cu. ft.}$$

Air Necessary for Combustion from Analysis of Fuel

Compounds

Owing to the uncertainty of the composition of hydrocarbon fuels, it is frequently convenient to determine the air necessary for combustion from an ultimate analysis of the fuel. This may be done as follows:

Having determined the percentage composition by weight, this percentage is expressed as a decimal when

$$\left(\frac{O_2}{C} \times \%C\right) + \left(\frac{O}{H_2} \times \%H\right) = \text{lbs. O per pound of fuel.} \quad (47)$$

Thus, an analysis of a standard brand of gasoline gave

$$C = 85.2 \text{ per cent.}$$
$$H = 14.8 \text{ per cent.}$$

Substituting these values in equation (47) we have

$$\left(\frac{32}{12} \times 0.852\right) + \left(\frac{16}{2} \times 0.148\right) = 3.45 \text{ lbs. of O}$$

and

$$\frac{3.45}{.236} = 14.6 \text{ lbs. of air lb. of fuel.}$$

Loss from Incomplete Combustion

Thermal Losses

It is thus seen that unless the air/gas ratio is at least 14.6, incomplete combustion will take place with its attendant loss. This loss is readily understood when we consider that 1 pound of carbon burned to CO_2 liberates 14,600 B.T.U., while 1 pound of carbon burned to CO liberates only 4,450 B.T.U., or but a little better than 30 per cent of the contained heat. The loss thus sustained is not in direct proportion to the CO present, as is sometimes stated, but is rather a function of the CO_2/CO

ratio. A convenient formula is given by Clerk & Burls for determining this loss. Slightly modified to use the lower heat value of the fuel this formula is

$$\frac{0.7}{1.03 + \frac{CO_2}{CO}} = \text{per cent of heat lost.} \qquad (48)$$

There are certain features of relative throttle opening reducing compression pressures, with resulting diminution of fuel efficiency

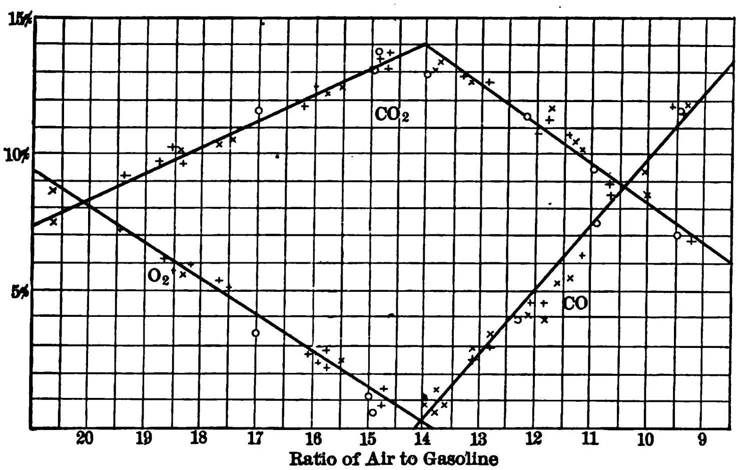


CHART XV.

Curves plotted from tests by Professor Watson, which show the relation between the products of combustion and the ratios of air to gasoline.

which compensate these figures slightly, but this quantity is quite negligible in the present consideration.

Chart XV shows the relationship of various air/gas ratios to the percentage of free O_2, CO_2, and CO in exhaust.

Dangerous Characteristics of Exhaust

Beside the inefficiency resulting from incomplete combustion, there are other disadvantages in having carbon monoxide in the exhaust gas. One serious consequence which may result under certain conditions is the possible poisoning of persons

who inhale the gas for a considerable length of time. CO is very poisonous when not diluted with other gases, and the effect is only less in degree when it forms but a comparatively small proportion of the gas inhaled. The evidence of poisoning may be nothing worse than a bad headache, but persons who work every day in ill-ventilated garages, the atmosphere of which is seldom free from the gas exhausted from motors, may easily suffer more serious consequences.

Imperfect combustion is also the cause of a foul smelling and often of a smoky exhaust. It is a well known fact that an over-rich mixture causes black smoke from this cause.

In cases where the exhaust leaving the motor contains both oxygen and CO as a result of poor mixing, the combustion may continue in the exhaust pipe and cause the latter to become excessively hot. This overheating often results in scorching the paint on parts adjacent to the exhaust pipe and may, under certain conditions, cause a serious fire. All of which are arguments in favor of securing the most complete combustion possible.

This overheating of the exhaust pipe may also be caused by the slow burning of a rich mixture which causes combustion to be continued after the exhaust valve has opened.

The same slow burning in a lean mixture causes combustion to be unfinished even upon the opening of the intake pipe which ignites the charge within the intake manifold, and causes back-firing, or "popping" as it is called, from the openings of the carbureter. Danger of fire from this phenomenon can be eliminated by placing gauze over the openings.

Aside from these considerations, the composition of the exhaust may be taken to indicate certain very definite conditions of carburetion. The following rules have been laid down and can be followed without error:

Economic Character of Exhaust

I. If the exhaust contains both CO and O_2 in considerable quantities (say more than 1 per cent of each) the presumption is that the gasoline and air were not well mixed, either because

of inadequate spraying (deposition) or insufficient heat for vaporization.

II. If CO appears in the exhaust without more than a trace of O_2 being present, the mixture is too rich and the supply of gas should be cut down.

III. If the exhaust contains only a trace of either or both O_2 and CO, the balance being CO_2, the combustion is complete—or substantially so. Probably a slight increase in the air will decrease the gasoline per horse-power hour.

IV. If the exhaust is free from CO and contains more than 4 per cent of O_2, the mixture is too lean and more gas should be admitted. It should be noted that the curves, Chart XV, indicate that when the exhaust contains 4 per cent of O_2, 11 per cent of CO_2, and no CO, the ratio of air to gas is 17 to 1.

Determination of Air/Gas Ratio

One of the most important uses of exhaust gas analysis is for the determination of the relative proportions of fuel and air which are present in the mixture. That this can be very closely approximated from the composition of the exhaust gases seems well established. Dr. Watson's curves, shown in Chart XV, are available only when the exhaust consists of either CO_2 and O_2 or CO_2 and CO. When both O_2 and CO are present, the air/gas ratio may be determined by use of the formula given by Clerk and Burls in "The Gas, Petrol, and Oil Engine," Vol. II, page 632, as follows:

$$\text{Corrected } \frac{\text{Air}}{\text{Fuel}} \text{ ratio by weight} =$$

$$\frac{2.86\,N}{0.532\,N - 0.4\,CO - 2\,(CO_2 + O_2)} \tag{49}$$

In this formula the chemical symbols are used to represent the volume per cent of the gases, and the coëfficients are based upon an analysis of the fuel, which was, in the case cited, C = 85.2 per cent, H = 14.8 per cent.

Ballantyne's Constant

In determining the nitrogen by difference, account must be taken of the presence of free H and CH_4, which are not ordinarily determined. Ballantyne has shown, however, that these constituents bear a constant ratio to the percentage of CO present in the following proportions:

Per cent of free H = 0.36 per cent of CO.

Per cent of CH_4 = 0.12 per cent of CO.

On page 631 of the same volume are shown comparative results of the formula from which the foregoing is derived, with results of actual measurements by Dr. Watson. The agreement is sufficiently close for all practical purposes, particularly if a numerator of 2.7 N is used when the ratio is 10 to 1 or less.

Importance of Air/Gas Ratios

The importance of the air/gas ratio is emphasized in some tests recently conducted by the Automobile Club of America.* Of this test three cars have been selected for purposes of illustration. All three cars were placed under strictly test conditions, so far as it was possible to place them on the road. Nine samples of exhaust gases were taken from each of the cars. The conditions of motor performance during the taking of the samples on the cars were as follows:

1. Car standing after motor had been running (motor running at low speed).
2. When car was accelerating to 10 miles per hour from stand.
3. Car running 10 miles per hour on level on second or third speed.
4. Car running 15 miles per hour on level on top speed.
5. Car running 20 miles per hour on level on top speed.
6. Car running 30 miles per hour on level on top speed.
7. Car climbing 6 per cent gradient at 20 miles per hour on top speed.
8. Car climbing 5.75 per cent gradient at 20 miles per hour on second or third.

* *The Automobile*, Feb. 12, 1914.

9. Car climbing 12.5 per cent gradient at 20 miles per hour on second or third.

The road surface while taking samples 3 to 6 was wooden block. Sample 7 was taken on oiled macadam, 8 on smooth wood block, and 9 while travelling over Belgian block.

The results of analyses are as follows:

Test No.	Car No. 1			Car No. 2			Car No. 3		
	CO_2	CO	O_2	CO_2	CO	O_2	CO_2	CO	O_2
1	11.1	2.4	0.4	9.1	4.4	1.0	5.2	5.1	4.2
2	13.6	0.0	0.6	11.1	2.4	0.3	7.0	4.6	0.9
3	13.	2.5	0.3	10.6	2.5	0.2	5.8	5.7	0.9
4	8.6	2.6	0.6	...	...	...	5.3	6.8	0.5
5	13.8	1.5	0.4	9.2	4.2	0.2	7.6	4.3	0.4
6	11.8	0.1	0.2	8.0	4.3	0.4	6.8	6.1	0.6
7	12.9	1.0	0.0	9.9	3.6	0.3	7.4	4.7	0.4
8	13.	0.8	0.1	6.6	6.8	0.4	8.3	3.6	0.5
9	11.8	2.0	0.2	6.4	6.8	0.4	7.2	4.4	0.8

Applying formula (49) to these analyses we find remarkable variations in the mixtures, not only in different cars but in the same car under different test conditions.

These results are plotted in Chart XVI, where test numbers are plotted as abscissæ with air/gas ratios as ordinates.

Maximum Power and Maximum Thermal Efficiency

Royal Automobile Club Standard

Dr. Watson (Proceedings I. A. E., Vol. III, page 405) has determined that maximum power is developed with an air/gas ratio of from about 11 to 13, while maximum thermal efficiency occurs with a ratio of about 17. Hopkinson and Morse (*ibid.*, 284), show that maximum thermal efficiency and maximum power occur practically together at a ratio of about 14. Experiments of the Massachusetts Institute of Technology show maximum power development with a ratio of about 12, which is in practical agreement with Dr. Watson's results. The Royal Automobile Club has decided the best mixture is at a ratio of 14.5 as giving from 90 to 95 per cent of both thermal efficiency

and maximum power. This is in reasonable accord with the determinations above cited.

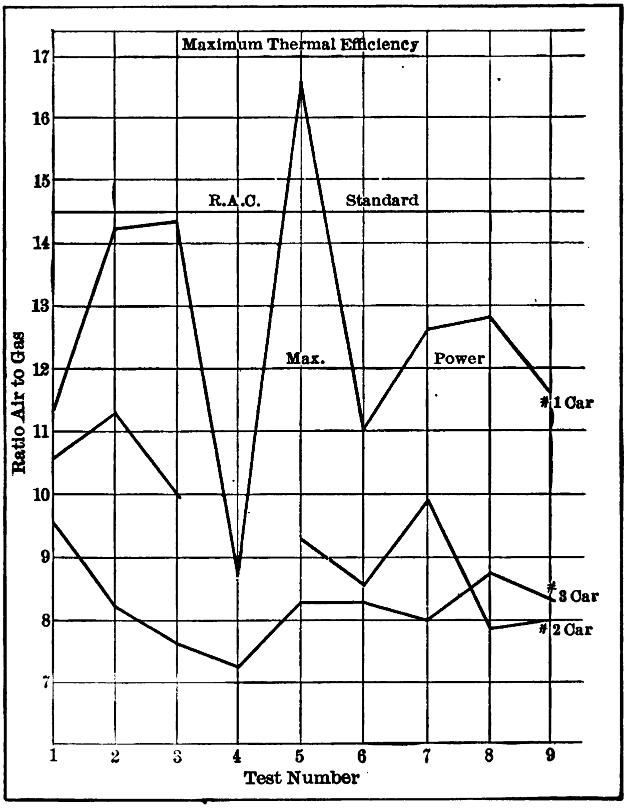


CHART XVI.

Advantages of a Constant Mixture

Complete combustion is possible only in the presence of *sufficient* air in *intimate admixture* with the fuel. Maximum pressures are obtainable only within very narrow limits of mixture composition. Both are essential to efficiency.

Chart XVII is plotted from a tabulation of experiments of the Massachusetts Institute of Technology. The time in seconds required for the explosion pressure to reach its maximum

is plotted against air/gas ratios by weight. The maximum pressure in pounds per square inch appears against each point.

Inspection of this chart shows that the greatest pressures are obtained when the rate of burning is fastest, and that departure

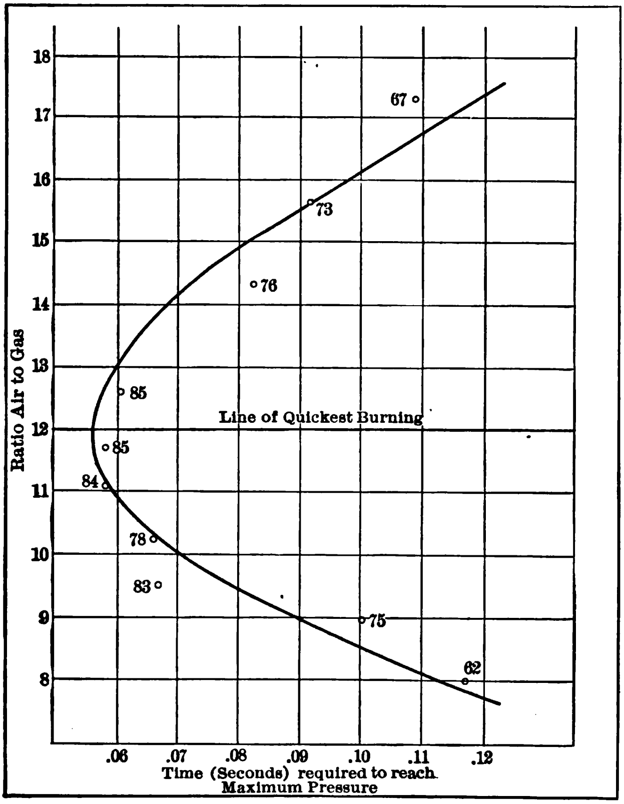


CHART XVII.

from the line of quickest burning either toward richness or leanness means a rapid falling off in power. It is true that by a proper spark advance this loss of power may be compensated for to some extent, but even an automatic spark control would have to be nimble to follow the varying ratios shown in Chart XVI.

Relative Volumes of Exhaust

When an excess of air is present in the mixture with gasoline vapor there is but a small increase in the final volume of the exhaust, but when the fuel is in excess the volume increase is

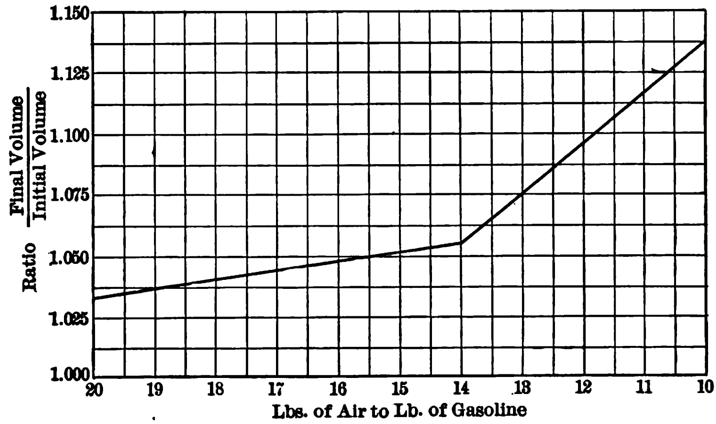


CHART XVIII.

quite large. The graph of Chart XVIII, from Dr. Watson (Cantor Lectures, 1910), shows this increase in volume ratios.

Rich Mixtures

It will be noted that with air/gas ratios from 14 to the limit of combustibility, the change is small, but that from 14 down the increase is comparatively rapid. Thus with a 10 to 1 mixture, the volume of the exhaust, reduced to the original temperature and pressure, would have increased nearly 14 per cent, indicating a substantial gain in power from this cause, but at the expense of a loss of heat shown in equation (48).

Furthermore, the gain by increased volume is offset by the reduction of explosion pressures of rich mixtures, as shown in Chart XVII. In some engine designs there undoubtedly is a final small gain in maximum power output afforded by enriched mixtures, provided ignition can be properly timed. It is doubtful however, if any pronounced accelerative effect is commonly

produced by sudden over-enrichment of the mixture because a favorable combination of the foregoing conditions is of rare occurrence.

This common misconception doubtless arises from the fact that many carbureters have a tendency toward impoverishment of the mixture upon the sudden opening of the throttle. In such a device it is probable that the actual "enrichment for acceleration" is not really as greatly in excess of the normal mixture as is commonly supposed.

Lean Mixture

On the other hand, a lean mixture entails similar losses without a proportional compensation of increased volume. Hence it is seen that cutting down the fuel does not necessarily mean economy, because, owing to reduced pressure and volume, a greater quantity of mixture is necessary to obtain a given road speed.

The lines of maximum thermal efficiency, maximum power, with some sacrifice of fuel economy, and the Royal Automobile Club standard, have been plotted in Chart XVI, and even casual inspection of the diagram will show how far the cars under test departed from ideal conditions. The reason for the relative fuel mileages of the cars is also apparent. The diagram also shows the erratic carbureter action to which the engines were subjected.

For instance, had the carbureter of Car No. 1 maintained throughout the test anything approaching the constancy it exhibited in tests 2 and 3, its fuel record, already good, would have been greatly improved. Had the carbureter of Car No. 3 maintained its same constancy with decreased fuel, this car would have, in all likelihood, surpassed the performance of Car No. 1 both in fuel mileage and general smoothness of operation.

Loss From Imperfect Combustion

Applying formula (48) to the analyses, we find fuel losses as direct as if the gasoline tank had been opened and its contents allowed to waste, as follows:

Car No. 1—6.7 per cent loss.

Car No. 2—23.0 per cent loss.
Car No. 3—30.0 per cent loss.

Free O_2 and CO

The foregoing losses are the direct result of the presence of CO and exist *because of it.* CO furthermore is, ordinarily, an *indication* of an over-rich mixture with all the losses that condition entails. This is not always the case, however, particularly when CO is present in small quantities, and even more obviously when it is associated with free O_2. The latter condition has excited much scientific speculation. That it is due largely, but not wholly, to imperfect contact of the molecules of fuel and air, the writer has demonstrated to his entire satisfaction. Claims have been made that liquid fuel particles actually passed through the cylinders unburned or but partially burned. This is more difficult of credence, but not impossible. There is, however, one theory which seems to have been generally overlooked, but which, if it is ever established, will demand serious consideration on the part of the designer. Some years ago MM. Mallard and LeChatelier demonstrated in a glass container that during a certain phase of concussive flame propagation, the flame was extinguished before combustion was complete. This they attributed to an action not unlike the echo of a sound wave. The vibratory character of flame propagation through an explosive mixture is commonly accepted, and it would seem possible that only a slight accentuation would be necessary to cause vibrations which might extinguish the flame. A careful study of this phenomenon might lead to distinct progress.

However, there is at present no better way of obtaining knowledge of the thermal and chemical reactions taking place within the gas engine cylinder than through the medium of exhaust gas analysis. That the results of gas analyses have seemed inconsistent at times is due rather to improper interpretation of results than to any inherent fault in the results themselves.

A Method of Analysis

Sampling

One of the chief reasons for apparently erratic results is that the analysis is performed upon samples not fairly representative

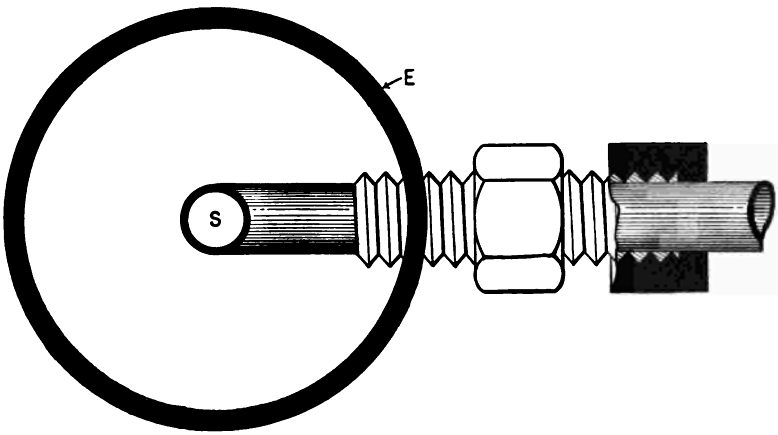


Fig. 19.

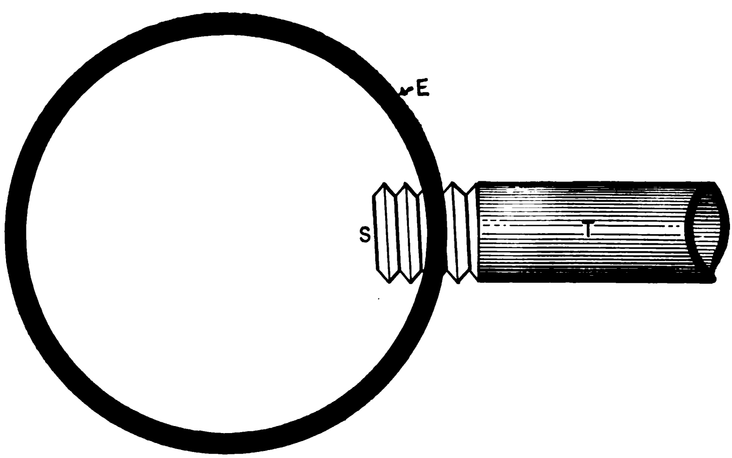


Fig. 20.

of the true composition of the exhaust gas. Accuracy depends fully as much on the method of taking the sample as upon the detail of the analysis itself.

The method of inserting a tube into the discharge end of the exhaust pipe is not to be countenanced. Between the pulsations of the exhaust is a period of diminished or even sometimes sub-atmospheric pressure. When this exists, air may be actually drawn into the exhaust pipe for a considerable distance, with consequent vitiation of any sample taken under these conditions.

The method of tapping a sample pipe into the exhaust pipe between the engine and the muffler is also inaccurate, even though the inner end of the sample tube be directed against the exhaust pressures. With such a device the sample is taken only of the center, or core, of the exhaust stream, as shown diagrammatically at S, Fig. 19, or if the pipe be not bent, only the surface of the stream will be sampled as at S, Fig. 20.

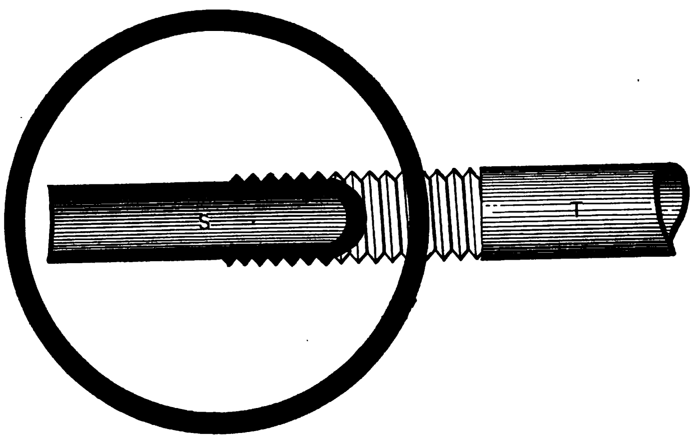


FIG. 21.

The most accurate location of the sample pipe seems to be as shown diagrammatically in Fig. 21.

In this arrangement the sample pipe T is inserted through the entire diameter of the exhaust pipe E. The portion within the exhaust pipe S is cut longitudinally so as to present an opening toward the flowing stream, across the entire diameter of the exhaust pipe. A sample withdrawn through this arrangement is fairly representative of the *entire stream* of the exhaust. Its

use will be found to give invariably consistent results, provided other conditions are properly met.

Leaky Exhaust Pipes

The exhaust pipes of very few cars will be found wholly free from air leaks, due to defective gaskets, poor threads, or even piping rendered porous by rust. The slightest leak is, of course, fatal to the accuracy of the sample. An excellent method to determine whether an exhaust line is tight is to set the carbureter adjustments so that far too rich a mixture is delivered to the cylinders. If the analysis of the exhaust shows even ½ per cent of free oxygen, it is conclusive proof that there is an air leakage and the same should be corrected before proceeding further.

Collecting the Sample

Having determined freedom from air leaks, the sample is taken by connecting the collecting tube, Fig. 22, with the sample pipe by means of a rubber tube.

The collecting tube is first filled with water, slightly acidulated with sulphuric acid to prevent the possible existence

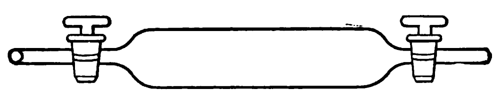


FIG. 22. GAS COLLECTING TUBE.

of free alkalies which would also absorb part of the CO_2 of the collecting sample. The collecting tube is then held in a vertical position and the upper stop-cock opened wide. The lower stop-cock is then opened, allowing the exhaust gas to replace the water which flows to waste.

ACTUAL ANALYSIS

The desired number of samples having been collected, each in turn is next transferred to an Orsat apparatus, Fig. 23.

The burette *P* holds 100 cubic centimeters, and is graduated in 1/10 c.c. It is filled with distilled water by elevating the bottle *B*. One end of the collecting tube is connected by means of a rubber tube to *A*. The other end of the collecting tube is connected to the water supply under slight pressure. The

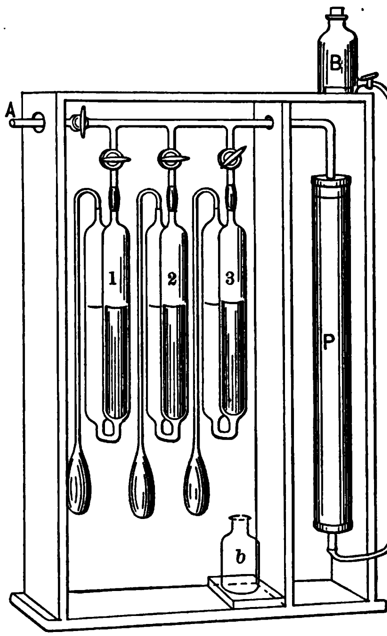


FIG. 23. ORSAT APPARATUS.

bottle *B* is placed below the burette as at *B*, and the stop-cock on the tube is opened. Upon opening the stop-cocks on the connecting tubes the gas will flow into the burette *P*, displacing the water therein into the bottle *B*.

More than 100 c.c. are so withdrawn, all stop-cocks are closed, and the collecting tube disconnected.

The cock *A* is now opened and the level carefully brought to zero by slightly elevating the bottle *B* and opening its stop-cock.

When zero is reached, exactly 100 c.c. of gas will be contained in the burette *P*, and the cock on *A* should be closed.

Determination of CO_2 and O_2

The pipette No. 1 contains a solution of potassium hydrate KOH. Into this the entire volume of gas is now passed by opening the stop-cocks on pipette and bottle, and elevating the latter. The gas is passed in and out of this several times and finally withdrawn into *P* until the KOH solution in No. 1 stands at the mark on the neck where it stood originally. It will now be found that the water in *P* no longer stands at zero. This is because the KOH solution has absorbed the CO_2 present. Consequently the reading of the burette is the volume percentage of CO_2 that has been absorbed. Before taking this reading, one minute should be allowed for all drainage to take place from the walls of the apparatus, otherwise the reading may be too low. Before any reading is accepted as final, the process should be repeated until two coincident readings are obtained. When this occurs the manipulation is repeated, this time passing the gas into pipette No. 2, which contains a solution of potassium pyrogallate. This solution absorbs oxygen, and its percentage is read on the burette as before, first deducting the CO_2 previously determined from the total burette reading.

Determination of CO

Again the process is repeated, this time into burette No. 3, containing a solution of cuprous chloride, which absorbs CO. This completed, the gas in the burette consists mainly of nitrogen, with small percentages of free hydrogen and marsh gas. There being no convenient method of determining these gases, we are obliged to estimate their percentage by Ballantyne's constant, noted in connection with equation (49). Their quantity is small and their constancy to the amount of CO present sufficiently permanent, so that the possible error caused by their non-determination is negligible.

It should be noted that while the absorption of CO_2 is very rapid, the second reading usually checking the first, the absorption of O_2 and CO is much slower, requiring many transfers for its accomplishment. With the average gas, a complete and accurate analysis usually requires about forty minutes.

CHAPTER VII

THE PHYSICAL CONDITIONS OF CARBURETION

NO LESS important than the chemical composition of a mixture is its physical condition. Certain aspects of this subject have been treated of in the chapter on Intake Manifolds, but its importance warrants a more thorough study.

As has been shown, the functions of carburetion are dual. Not only must the fuel be mixed with a definite amount of air, but, to be effective, the fuel must be absorbed by the air. Air being a gas, no absorption of fuel can take place until it, too, becomes a gas.

HEAT

This involves the absorption of heat for two purposes: First, to raise the temperature of the fuel to the evaporation point; and second, to supply the heat absorbed by actual vaporization.

Specific Heat

The British thermal units necessary to raise the temperature of one pound of a substance one degree Fahrenheit is called the *Specific Heat of the Substance.*

Latent Heat

The heat absorbed without change of temperature during a change of state, as from a solid to a liquid or from a liquid to a gas, is called the latent heat.

Gasoline, as it is called in this country, is of uncertain chemical composition, and its specific heat and latent heat are therefore uncertain. We quote the following from leading authorities:

TABLE II.

TABLE OF LATENT AND SPECIFIC HEATS OF PETROLEUM PRODUCTS

Product	Specific Heat	Temperature	Latent Heat	Authority
Petroleum	0.511	21° to 58° C	.....	Pagliana
Petroleum	0.498	18° to 99° C	.....	Pagliana
Crude Petroleum, Japan	0.453	....	.....	Mabery and Goldstein
Crude Petroleum, Penn	0.500	....	.....	Goldstein
Crude Petroleum, California	0.398	....	.....	Goldstein
Crude Petroleum, Russia	0.453	....	.....	Goldstein
Kerosene Sp. Gr. .810	0.499	....	105.4	Redwood
Kerosene Sp. Gr. .811	0.470	260° F*	.....	Robinson
Naphtha, Sp. Gr. .756	0.510	175° F*	103.5	Redwood
Naphtha, Sp. Gr. .720	0.569	115° F*	100.6	Redwood
Gasoline, Sp. Gr. .642	0.580	70° F*	100.2	Redwood
Petroleum ether	0.445	100° C	.....	Eckerlein
Petroleum ether	0.419	0° C	.....	Eckerlein
Benzol (C_6H_6)	0.407	10° C	.....	Pickering
Benzol (C_6H_6)	0.450	50° C	.....	Pickering
Benzol (C_6H_6)	0.482	65° C	.....	Deruyts
Benzol (C_6H_6)	.....	0° C	109.	Regnault
Benzol (C_6H_6)	.....	80.1 C	92.9	Wirtz
Benzol (C_6H_6)	.....	80.35C	93.5	Schiff

* Boiling-points.

From the foregoing table it is seen that averages, sufficiently close for all practical purposes, may be assumed to be

Specific heat = 0.500 B.T.U.
Latent heat = 100.0 B.T.U.

Specific and Latent Heat of Gasoline

On no subject connected with gasoline as a fuel does there seem to be such a divergence of views as upon the latent heat. Table II is a compilation from authorities quoted by the U. S. Bureau of Standards, supplemented by other recent authorities.

Total Heat

Latent heat must not be confused with *Total Heat*, which is the heat necessary to raise the temperature of a substance to a given degree, plus the latent heat of vaporization at that temperature.

The subject of the specific and latent heats of fuel is of primary importance in practical carbureter design. It is highly

probable that the lack of definite knowledge concerning these factors has seriously retarded the use of heavier fuels. Furthermore, a more complete grasp may be obtained of problems of starting, water-jacketing, and the like by a thorough understanding of the action of heat.

Reduction of Temperature by Evaporation

Considering first the temperature drop occasioned by the evaporation of gasoline, let us assume that the carbureter is delivering a mixture of 1 part of gasoline to 15 parts of air by weight. Let us assume the temperature of the air, and consequently that of the gasoline, to be 60° F.

The specific heat of air at constant pressure is 0.2375, therefore the total heat available for each degree drop in temperature is $(1 \times .500) + (15 \times .2375) = 3.0625$ B.T.U. But as the heat necessary to vaporize 1 pound of gasoline is 100 B.T.U., the resulting temperature drop will be

$$\frac{100}{3.06} = 32.4^\circ$$

and the resulting temperature in the carbureter or manifold will be

$$60 - 32.4 = 27.6^\circ \text{ F.}$$

Frost-Covered Manifolds

This accounts for the appearance of frost on the manifold until the temperature beneath the hood raises sufficiently to supply the necessary heat by the conductivity of the walls of the manifold.

Effect of Mixture Proportion

The richer the mixture, the greater the temperature drop, provided the fuel is vaporized, and the fact that some manifolds do not indicate this marked temperature drop is proof that a large percentage of the fuel is carried to the cylinder still in the form of a liquid.

Every liquid fuel has a definite temperature below which no inflammable vapor is given off. With the commercial gasoline of the present day, this temperature is increasing as the gravity decreases. It is too uncertain to fix a definite value, but the difficulty of starting a cold engine is attributable directly to the reduction of temperature within the carbureter below this critical point.

Flash-Point

An illustration of this is furnished by an attempt to start with kerosene mixed with gasoline. The vaporization temperature, or "flash-point," of kerosene is upward of 80° F. If the temperature drop is 32°, as noted, the initial temperature of both air and fuel must be at least 80 + 32, or 112° F., for complete evaporation. This is recognized in kerosene carbureters, which commonly start the engine on the more volatile gasoline, continuing the operation with kerosene only when the necessary heat has been supplied by the engine.

Necessity for Artificial Heat

In a 15 to 1 mixture

$$15 \times .2375 = 2.56 \text{ B.T.U.}$$

must be supplied to the air for each pound degree rise of temperature, while .5 B.T.U. is supplied to the fuel. Consequently, in the use of heavier fuels at least, the necessity for pre-heating the air is apparent.

Loss of Volumetric Efficiency by Heat

It is frequently urged against this practice that the weight of mixture entering the cylinder is decreased by heat with consequent loss of power. The extent of this loss of volumetric efficiency may be determined by a consideration of the expansion of gases by heat.

According to the law of perfect gases

$$\frac{PV}{T} = R, \text{ a constant,}$$

when

$$P = \text{pressure.}$$
$$V = \text{volume.}$$
$$T = \text{absolute temperature.}$$

At 32° F. a cubic foot of dry air at sea level weighs 0.080728 pounds. The volume of 1 pound is, therefore, $\frac{1}{.080728} = 12.387$ cubic feet. The pressure per square foot is 2116.2 pounds.

$$\frac{PV}{T} = \frac{2116.2 \times 12.387}{491.13} = 53.37$$

If the temperature of a gas is raised 1° F. (from 32° to 33° F.), we find that the volume of 1 pound is

$$\frac{RT}{P} = V \quad \frac{53.37 \times 492.13}{2116.2} = 12.411 \text{ cubic feet.}$$

That is, for a rise in temperature of each 1° F., the volume of a given weight will be increased

$$\frac{12.387}{12.411} = .00196$$

or practically $^2/_{10}$ of 1 per cent.

If, therefore, air is admitted to the carbureter at, say 100° F. above the atmospheric temperature, the volumetric loss from heating would be about

$$0.2\ (100^\circ - 32.4^\circ) = 13.5 \text{ per cent}$$

when all the fuel is vaporized before reaching the cylinder.

Partial Vaporization

It is evident that if but partial vaporization takes place the reduction of the temperature will be less, and the volumetric loss proportionately more. This is a strong argument in favor of complete evaporation of the fuel *before* it reaches the cylinder of a four-stroke cycle engine. With the two-stroke cycle, evaporation is usually completed by the temperature and mechanical

agitation within the crank case, so that the charge is delivered to the combustion chamber at fully reduced temperature.

Effect of Vaporization Within the Cylinder

It has been stated that with some four-stroke cycle marine engines, evaporation of liquid fuel particles within the cylinder during the compression stroke causes an increase of power by a reduction of the energy expended in compression. The evidence submitted points to the accuracy of this observation, but it is doubtful if it is the direct result of an actual reduction of temperature. Lower temperature means lower compression pressure, which in turn implies reduced efficiency and power output. It seems probable that, if the claim is correct, it is due to heat transference from the hot walls, restoring or even increasing the normal compression temperature.

Economy of Artificial Heat

Certain it is that heat is desirable with the present grades of fuel. This is indicated by the greater mileage secured by recent systems of fuel feed, wherein an appreciable quantity of fuel is held in a sheet metal reservoir in proximity to, and receiving the heat from, the engine. The heat necessary to raise the fuel to its evaporation point, and also at least a portion of the heat necessary for evaporation, are secured in this way. Under these conditions, vaporization is more complete and greater efficiency is the direct result.

STARTING

In starting an engine when none but atmospheric heat is available, but one alternative remains when temperatures are low. A fuel mist must be introduced into the cylinders and these liquid particles ignited and burned until the temperature raises sufficiently to supply heat for the gasification of the fuel, either from a water-jacket, heated air from around the exhaust pipe, or by radiation and conduction from the engine itself.

"Rich Mixture for Starting"

Such an understanding of actual conditions shows the utter fallacy of the oft reiterated statement that "a *rich mixture* is necessary for starting." The fact is that under ordinary conditions, no *gaseous* mixture whatever is delivered to the cylinders, or at best but a mixture containing so little *fuel gas* as to be uninflammable. Such portion of the liquid fuel as is not deposited in carbureter and manifold reaches the combustion chamber as liquid particles more or less finely divided. In order to ignite at all, there must be far more of these liquid particles present than would be necessary were they evaporated. When a *true gas* fuel is delivered to the cylinders (as in the case of an engine using illuminating gas as a fuel) there is no trouble with starting at *any* temperature without changing the mixture proportions from those of continued running. Hence the statement that "a rich mixture is necessary" should become "an excess of liquid fuel is required for starting." Such an expression would help materially to remove one of the stumbling-blocks from the path of carbureter development.

Effect of Temperature on Fuel Flow

Viscosity of Gasoline

The property of viscosity is not ordinarily associated with liquids as light as gasoline. It is a fact, however, that the flow of gasoline through the nozzle of a carbureter is directly affected by changes of temperature. Chart XIX shows the results of measurements of flow at temperatures between 50° and 100° F. It will be noted that at 100° F. the discharge of a nozzle is about 36 per cent greater than at 50° F.

It is thus seen that a carbureter nozzle adjusted to give a proper mixture at a working temperature of 100° F. will discharge but a little over 71 per cent. of the requisite fuel when the temperature falls to 50° F., which, as has been shown, is the very time when an *excess* of fuel is needed.

Like other characteristics of gasoline, its viscosity varies with its composition, therefore definite regulation of the fuel orifice, thermostatic or otherwise, is consequently difficult.

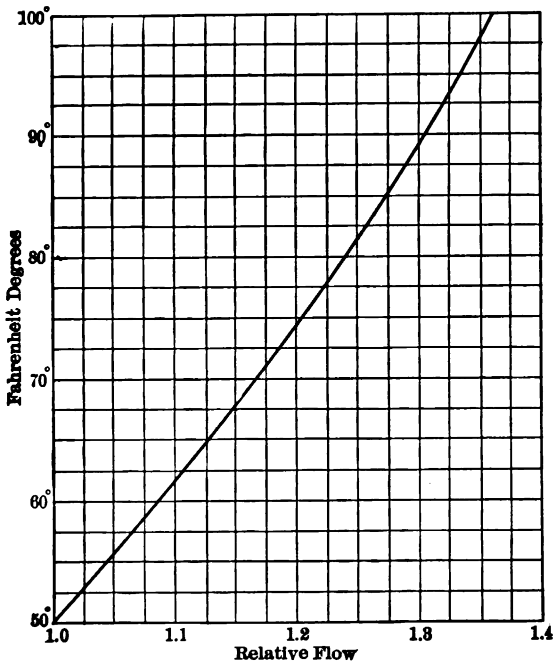


CHART XIX.

Pressure

Variation of Compression Pressures

At practical sea level, the normal pressure of the atmosphere is 29.92 inches of mercury, equivalent to 14.7 pounds per square inch. The maximum variation is about 3 inches, *i.e.*, from 28 to 31 inches, about 10 per cent. The density or weight of a cubic foot of air varies inversely as the pressure, hence when the barometer stands at 31 inches, 10 per cent more weight of mixture should be delivered to the cylinders than during periods of extreme low pressures.

Effect of Reduced Pressures on the Auxiliary Valve

What is commonly called the "vacuum" in a carbureter is really an expression of the pressure *difference* between the interior of the carbureter and the outside atmosphere. Lower pressures mean lower pressure differences, or, as it is commonly called, "less vacuum." This causes the auxiliary air-valve to open a lesser amount. Now at a given engine speed the same volume of air (at reduced density) is drawn into the cylinders and, as the air-valve is opened a lesser amount, the total admission area is decreased. This entails a higher velocity over the fuel jet. This increase of velocity is sufficiently high to more than compensate for the decreased density (see equation 5) and, as a result, too much fuel is inspirated. Thus, unless a carbureter is governed by velocity, independently of other pressure differences, it will be susceptible to barometric changes.

Effect of Altitude on Compression Pressures

This effect is particularly noticeable at higher altitudes when, if adjustments are not necessary, it is simply an indication that the engine had previously been running at lower altitudes or with an inefficient mixture.

Effect of Altitude on Vaporization

Chart XX shows the barometric pressure at different altitudes, and also the effect of the diminished pressure on the boiling-point of water. This evaporative effect of reduced pressures is, of course, even more pronounced on liquids of lighter gravity, such as gasoline, so that vaporization is more complete at higher altitudes. As noted in a previous paragraph, this occasions a correspondingly greater temperature drop, and hence winter starting at high altitudes is usually more difficult than at sea level, at the same temperature.

It is common to experience a noticeable lack of power at high altitudes. This may be due to no fault of the carbureter, after the latter is properly adjusted to meet the new conditions.

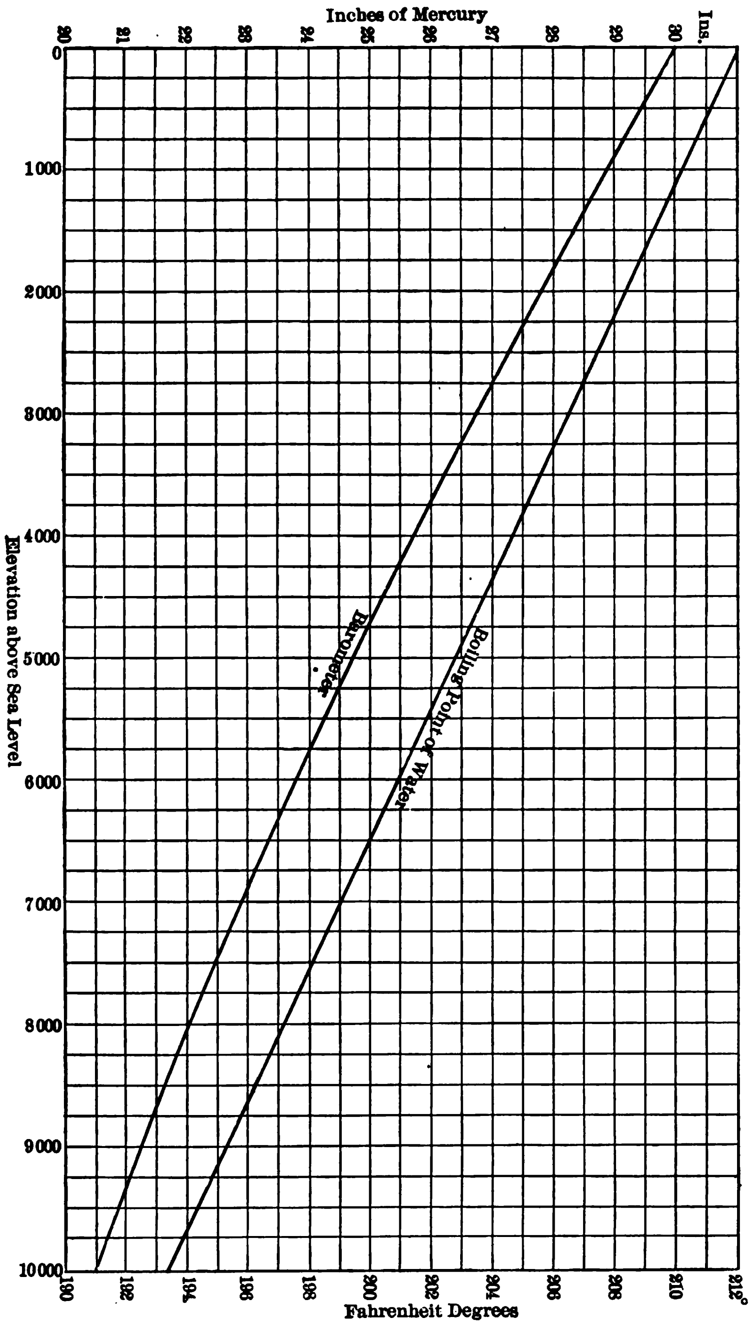

Inches of Mercury
20
21
22
23
24
25
26
27
28
29
30
Ins.
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
Elevation above Sea Level
Barometer
Boiling Point of Water
190
192
194
196
198
200
202
204
206
208
210
212°
Fahrenheit Degrees

CHART XX.

Air Standard of Efficiency

The efficiency of an internal combustion engine is a direct function of the compression. For any gas engine where

r = compression ratio,
Cp = specific heat at constant pressure,
Cv = specific heat at constant volume,
E = efficiency,

$$\gamma = \frac{Cp}{Cv}$$

and

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1},$$

which shows that high efficiencies depend upon high compression.

Effect of Altitude on Power

The volumetric efficiency of compression decreases approximately 3 per cent with each thousand feet of altitude, so that at 10,000 feet we find a reduction in compression of about 30 per cent. The loss of power entailed by this reduction in compression can be, even partially, compensated by the carbureter, only in the event that the mixture, originally too poor, becomes enriched from causes already mentioned. Because this combination of circumstances sometimes happens in touring, claims are set forth that the carbureter used is insusceptible to barometric changes. Such claims lead to confusion which only retards practical development. Usually, however, the effect of such an altitude is to so vitiate the proportions of the mixture as to make readjustments imperative. If this is made so that a slightly richer mixture than normal is obtained, it is possible to compensate for a considerable portion of the volumetric loss.

CHAPTER VIII

THE CARBURETER OF THE FUTURE

Balanced Forces

CAREFUL study of the problems of carburetion indicates certain definite relations between the various forces employed. The ideal carbureter, if it ever arrives, will express the delicate balance between these forces in its design.

Changing Fuel

The chief difficulty to be surmounted lies in the uncertain and ever-varying composition of the fuel. It has been true in the past that the most efficient carbureter would become relatively useless owing to changes in composition of commercial fuel. There are good reasons for believing that this condition exists to-day to a far less extent. A carbureter adapted to handle the fuels of the present day will employ principles which will be found commercially operative with lower grades of fuel than are likely to be forced into general use for many years.

Constancy of Mixture

The ideal carbureter design at present indicated seems to necessitate the embodiment of ten salient features.

1. Mixture composition must be automatically maintained at any desired air/gas ratio, irrespective of speed changes, load or weather conditions. Neither accelerating nor idling should change the mixture proportions, but if such constancy is unattainable the tendency should be toward enrichment rather than impoverishment of the mixture.

Atomization

2. High velocities must be employed, at least over the fuel nozzle, in order to produce the fine atomization necessary for efficient vaporization.

Velocities

Particularly should the velocity at cranking speed be high. At this time no heat is available for vaporization, and consequently whatever fuel gas *is* found must be the direct result of decreased pressures acting on the greatest surface possible, a condition synonymous with high velocity.

At high speeds the velocity may, in fact should, decrease in order to obtain the extreme limit of volumetric efficiency. At high speeds much heat is generated and this may be utilized to produce the necessary gasification of the fuel.

Whether the jet should assume the form of a single nozzle or many, a slot, an annulus, or other form, is a matter for experimental determination.

Size and Shape of Passages

3. Passages for the flow of both air and gasoline should conform, so far as possible, to the natural shape of the *vena contracta*. The venturi-shaped passage makes for constancy of the coëfficient of flow, and therefore permits of more accurate determination of dimensions. Unnecessary bends should be avoided, particularly in all air passages, and sharp corners or baffling projections should not be tolerated. Passages should be made with sufficient cross-sectional area not to exert undue wire-drawing, or throttling of the charge. In brief, the internal resistance of the instrument should be at the minimum.

Application of Heat

4. The application of the heat to the instrument should be made with full recognition of the thermal reactions involved. If error is made, it should be toward the side of excess heat, for, notwithstanding the loss of volumetric efficiency involved, this is always partially and sometimes wholly overcome by the greater rate of flame propagation, with the consequent rise of explosion pressure, which is induced by a higher initial temper-

ature of the charge. Here again must the *balance* be established in conformity, to a certain extent, with engine design.

Water-Jacketing

Lacking definite knowledge of conditions and still to design a universal carbureter, it is well to provide means for regulating the temperature of the heated air, as by an adjustable vent in the hot air pipe. Regulation of the temperature of jacket water is less necessary, because, in ordinary design, the passage of the air through the carbureter is of such short duration that little heat is absorbed by the air. The water-jacket serves only to supply heat for the vaporization of deposited liquid fuel and because of the reduction of temperature incident to this process, there is little danger of raising the temperature of the air unduly. The jacketing, however, should be confined to the walls of such passages as cause deposition by their enlargement of the cross-sectional area and consequent reduction of the velocity.

Adjustments

5. Adjustments, however, should be minimized, if not abolished altogether. It will take many years of education to convince the truck-driver that he does not know more about his carbureter than the designer, and until that Utopian day arrives the engineer can protect himself, his reputation, and his product only by removing every adjustment which he considers not absolutely vital.

Fuel Level

What adjustments are essential will depend upon the type and design of the instrument, but it seems certain that many of these at present in use can be abolished. For instance, inexperienced operators seem to delight in readjusting the level in the fuel reservoir, notwithstanding that its practical effects on the mixture are wholly negligible. By equation (13) it may readily be determined that a difference of even one-half inch gives rise to an inappreciable error even at the highest velocities.

Moving Parts

6. Moving parts should be abolished as far as possible. What remain should be so constructed as to not change their functions even after the normal wear to which every mechanism is subject.

No moving part should have a sliding fit in the direction of the air-current. Dust and dirt are liable to render such a device inoperative. Undue wear, or even constant wear, must be provided against, and to this end movements should be small in extent and intermittent in character.

Accessibility

7. Accessibility should be a prime consideration. The design should be such that complete disassembly can take place without removing the carbureter from the manifold. The gasoline nozzle and all its passages should be particularly accessible, for, despite any filtering devices, dirt at times persists in selecting the fuel nozzle as a resting place. The float reservoir, too, should be specially accessible and should be provided with a convenient drain whereby an accumulation of water and other impurities may be occasionally removed.

Priming

8. The carbureter should be provided with some device for supplying an excess of fuel when cold. This device should be automatic in its action but must be wholly inoperative except at relatively low temperatures, otherwise it would function every time the engine was slowed to cranking speed. The result would be too rich a mixture at low speed, entailing a useless waste of fuel, carbon deposits in the cylinder, and general unsatisfactory action.

Fire Protection

9. If means are ever found whereby the fuel flow is automatically compensated for temperature, danger from fire will be

practically unknown. Until then, openings should be protected with wire screen sufficiently heavy to afford the necessary cooling but not of so fine a mesh as to become readily clogged and thus prevent the admission of the proper amount of air.

Practical Manufacture

10. Finally, the whole must be embraced in a design capable of the most advanced manufacturing methods. Interchangeability of parts is an intensely practical requirement to manufacturer and user alike: to the first, because of reduced manufacturing cost; to the latter, because in the event of accident he is certain of prompt replacement.

Summary

Summarized, there is no reason why the carbureter should not become as standard and reliable a product as the engine itself. Its functions are in reality far less involved, and the avowed idiosyncrasies of the carbureter of to-day have existence only in our lack of knowledge concerning the principles of carburetion. The day cannot be far distant when an efficient combination of capital and engineering skill will solve the remaining problems, thereby increasing automobile efficiency in the broadest sense of the term, by bringing carbureter troubles to an end.

APPENDIX

USEFUL TABLES AND CONVENIENT FORMULÆ

1. Vacuum Gauge

To Find Absolute Pressures Shown by

Let R = vacuum gauge reading,
B = barometer reading,
P = absolute pressure,

then $$(B - R)\ .49 = P.$$

2. Mercury Columns

1 inch mercury weighs .49131 pounds per square inch. (Log 1.6913557.)

1 inch mercury = 13.647 inch H_2O. (Log 1.1350532.)

3. Compression Efficiency at Different Altitudes

TABLE III.

For air at 70-pound gauge pressure. (Hiscox.)

Feet Above Sea Level	Volumetric Efficiency of Compression, Per Cent	Loss in Capacity, Per Cent	Decreased Power Required, Per Cent
0	100.	0.	0.
1000	97.	3.	1.8
2000	93.	7.	3.5
3000	90.	10.	5.2
4000	87.	13.	6.9
5000	84.	16.	8.5
6000	81.	19.	10.1
7000	78.	22.	11.6
8000	76.	24.	13.1
9000	73.	27.	14.6
10000	70.	30.	16.1
11000	68.	32.	17.6
12000	65.	35.	19.1
13000	63.	37.	20.6
14000	60.	40.	22.1
15000	58.	42.	23.5

Note.—For pressures above 70 pounds gauge, deduct 3 per cent from the figures in column 2 above, and 10 per cent from the figures in the last column, for each 10 pounds increase above 70 pounds (approximate).

4. To Reduce Head in Feet to Pressure in Pounds per Square Inch

Let p = pressure of atmosphere in pounds per square inch.
P = pressure of atmosphere in pounds per square foot = $144p$.
H = head in feet necessary to cause atmospheric pressure, p.
h = head in feet necessary to cause pressure of 1 pound.
Wa = weight of 1 cubic foot of air.

then $$\frac{P}{Wa} = H$$

and $$\frac{H}{p} = h$$

or $$\frac{p}{H} = \text{pounds per square inch for 1-foot head.}$$

Thus, if $p = 14.7$

$Wa = .076$

$$\frac{14.7 \times 144}{.076} = 27816$$

and $$\frac{27816}{14.7} = 1892.22$$

or $$\frac{14.7}{27816} = .00052847 \text{ pounds per square inch for 1-foot head.}$$

5. To Determine Drop in Pressure by Velocity

Let

h = head in feet to cause 1 pound pressure $= \frac{27816}{14.7} = 1892.22$ ft.

v = velocity in feet per second.
p = drop in pressure.

$$p = \frac{v^2}{2\,gh}$$

6. Velocity of Flow

Let

v = velocity of flow in feet per second.
p = pressure causing the flow in pounds per square inch.
g = acceleration of gravity 32.2 feet per second per second.
h = head in feet necessary to cause a pressure of 1 pound = 1892.22 feet.
c = coëfficient of flow.

then $$p = \frac{v^2}{2\,gh}$$

consequently $$v^2 = 2ghp \times c$$

whence $$v = c\sqrt{2\,gh}\,\sqrt{p}$$

or $$v = 348.87\,c\sqrt{p}$$

7. Manometer Pressures

1 inch water = .036 pounds.
1 inch mercury = .491 pounds.
1 inch mercury = 13.647
27.8 inch water = 1 pound.
38.6 inch gasoline specific gravity 0.72 = 1 pound.
2.4 inch mercury = 1 pound.

8. Displacement

Let

D = displacement of cylinders in cubic inches.
Ve = volumetric efficiency.
g = gear ratio.
M = miles per hour.
d = diameter drive-wheels in inches.
A = area.
s = stroke in inches.
n = number of cylinders.
b = bore in inches.

R = r.p.m. of engine.

$$D = b^2 .7854\, sn.$$

$$D = \frac{3456 \text{ cubic feet per minute}}{R \times Ve}$$

$$D = \frac{120 \text{ cubic inches per second}}{R \times Ve}$$

Engine speed, $R = \dfrac{336\, gM}{d}$

Cubic feet per revolution, $= \dfrac{DVe}{3456}$

Cubic feet per minute $= \dfrac{DRVe}{3456}$

Cubic inches per second $= \dfrac{DRVe}{120}$

Cubic inches per second $= \dfrac{2.8\, gMDVe}{d}$

Velocity in feet per second $= \dfrac{DRVe}{1440A}$

Velocity in feet per second $= \dfrac{DgMVe}{4.2857A}$

Car speed in miles per hour, $M = \dfrac{Rd}{336g}$

9. Miles per Hour to Feet per Second

$$\frac{5280}{60 \times 60} \times M.P.H. = \text{feet per second}$$

or

$$1.4666\ M.P.H. = \text{feet per second (Log } 0.1663304).$$

10. To Determine Volume Ratios from Weight Ratios

Let

Wf = weight 1 cubic foot fuel vapor.
A = air/fuel ratio by weight.
Wa = weight of 1 cubic foot of air.
R = air/fuel ratio by volume.

then

$$\frac{Wf \times A}{Wa} = R$$

and

$$\frac{1}{R} = \text{per cent of fuel by volume.}$$

11. Acceleration Computations

Let

v' = initial velocity in feet per second.
v'' = final velocity in feet per second.
a = acceleration in feet per second per second.
t = time in seconds.

then

$$v' = v'' - at$$

$$v'' = v' + at$$

$$a = \frac{v'' - v'}{t}$$

$$t = \frac{v'' - v'}{a}$$

12. Computations of Velocity

Cubic ft./min. $\times$ 28.8 = cubic inches per second (Log 1.4593925).

Velocity^2 in feet per second $\times$ .00022735 = inches water (Log 4.3567071).

Inches water $\times$ 4398.3 = velocity in ft./sec. (Log 3.6432929).

13. Loss of Pressure in Pipes (Kent)

Let

p = pressure loss in pounds per square inch.
v = velocity of air/feet per second.
L = length of pipe in feet.
d = diameter of pipe in inches.

$$p = 0.0000025 \frac{Lv^2}{d}$$

$$v = 632.5\sqrt{\frac{dp}{L}}$$

$$d = \frac{.0000025\ Lv^2}{p}$$

14. Effect of Bends (Kent)

Radius of bend in diameters of pipe,	5	3	2	1½	1¼	1	¾	½
Equivalent lengths of straight pipe diameters,	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

15. Water

Weighs 62.355 pounds per cubic foot.
1 foot head = 0.433 pounds per square inch.
1 inch head = 0.0360860 per square inch.
1 pound pressure = 2.306 feet head.
13.647 inches = 1 inch of mercury.

16. Volumetric Efficiency

$$\frac{\text{Actual cubic feet per minute}}{\text{Displacement in cubic feet per minute}} = \text{Volumetric efficiency.}$$

17. Brake Horse-Power

Let

S = r.p.m.
T = scale reading in pounds.
r = radius of brake-arm.

then

$$\frac{ST2\pi r}{33000} = \text{B.H.P.}$$

18. Capacity of Prony Brakes

Each square foot of rim surface of a water-cooled iron brake pulley will absorb 10 B.H.P. without undue heating.

19. Formulæ for Temperature Correction for Specific Gravity of Gasoline

Let

S = specific gravity at 60° F.
s = specific gravity at t° F.
t = temperature in F.°

then

$$S = \frac{s}{1 - .0007\,(t - 60)}$$

TABLE IV.

20. Weight of Gases at 32° F. and 29.92 Inches Mercury (Kent)

	Pounds per Cubic Feet	Cubic Feet per Pound
Air	0.080728	12.388
Hydrogen	0.00559	178.931
Oxygen	0.08921	11.209
Nitrogen	0.07831	12.770
Carbon monoxide	0.07807	12.810
Carbon dioxide	0.12267	8 152

TABLE V

21. Baumé's Hydrometer and Corresponding Specific Gravity (Kent)

Formula. Specific Gravity = 140 ÷ (130 + degrees B)

Degrees Baumé	Specific Gravity	Degrees Baumé	Specific Gravity
10.0	1.000	32.0	0.864
11.0	0.993	33.0	0.859
12.0	0.986	34.0	0.854
13.0	0.979	35.0	0.849
14.0	0.972	36.0	0.843
15.0	0.966	37.0	0.838
16.0	0.959	38.0	0.833
17.0	0.952	39.0	0.828
18.0	0.946	40.0	0.824
19.0	0.940	41.0	0.819
20.0	0.933	42.0	0.814
21.0	0.927	44.0	0.805
22.0	0.921	46.0	0.796
23.0	0.915	48.0	0.787
24.0	0.909	50.0	0.778
25.0	0.903	52.0	0.769
26.0	0.897	54.0	0.761
27.0	0.892	56.0	0.753
28.0	0.886	58.0	0.745
29.0	0.881	60.0	0.737
30.0	0.875	65.0	0.718
31.0	0.870	70.0	0.700
....	.....	75.0	0.683

22. British Thermal Unit

1 B.T.U. = the amount of heat necessary to raise 1 pound of water from 62° to 63° F.

Note.—It takes slightly more than 1 B.T.U. to raise 1 pound of water 1° F. above 63° F. and slightly less below 62° F., but these quantities are so small as to be negligible in practice.

1 calorie = 3968 B.T.U. = 1 kilog. of water 1° C.

1 B.T.U. = 778 foot-pounds of work = Joule's Equivalent.

42.42 B.T.U. per minute = 1 horse-power.

2545 B.T.U. per hour = 1 horse-power.

To Find B.T.U. Equivalent to any Rise in Temperature

$$\text{Rise in F.}^\circ \times \text{weight} \times S = \text{B.T.U.}$$

To Find the Rise in Temperature by the Addition of a Given Number of B.T.U.

$$\frac{\text{B.T.U.}}{\text{weight} \times S} = \text{Rise.}$$

when S = specific heat (*q. v.*).

TABLE VI

23. VOLUME, PRESSURE, AND DENSITY OF AIR

From a normal volume and pressure of 62° Fahr. (Haswell.)

F.	Volume of a Pound	Absolute Pressure of a Constant Volume of Heat	Density or Weight of 1 Cubic Foot of Free Air
0	11.583	12.96	.086331
32	12.387	13.86	.080728
40	12.586	14.08	.079439
50	12.840	14.36	.077884
62	13.141	14.70	.076097
70	13.342	14.92	.074950
80	13.593	15.21	.073565
90	13.845	15.49	.072230
100	14.096	15.77	.070942
120	14.592	16.33	.068500
140	15.100	16.89	.066221
160	15.603	17.50	.064088
180	16.106	18.02	.062090
200	16.606	18.58	.060210
210	16.860	18.86	.059313
212	16.910	18.92	.059135
220	17.111	19.14	.058442
240	17.612	19.70	.056774
260	18.116	20.27	.055200
280	18.621	20.83	.053710
300	19.121	21.39	.052297
320	19.624	21.95	.050959
340	20.126	22.51	.049686
360	20.630	23.08	.048476
380	21.131	23.64	.047323
400	21.634	24.20	.046223
425	22.262	24.90	.044920
450	22.890	25.61	.043686
475	23.518	26.31	.042520
500	24.146	27.01	.041414
525	24.775	27.71	.040364
550	25.403	28.42	.039365
575	26.031	29.12	.038415
600	26.659	29.82	.037510
650	27.915	31.23	.035822
700	29.171	32.635	.034280
800	31.681	35.445	.031561
900	34.197	38.255	.029242
1000	36.811	41.065	.027241
2000	61.940	69.165	.016172
3000	87.130	97.265	.011499

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